# PRODUCT ENGINEERING

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# *The Brain Supply*

IN ENGLAND the aristocracy of brains is found in the diplomatic service, in Continental Europe it is in the universities, while in the United States it is concentrated in the engineering and research laboratories of industry. This is the opinion of Sheldon and Martha Candler Cheney, authors of the recently published book, "Art and the Machine," and it is difficult to say wherein they are wrong.

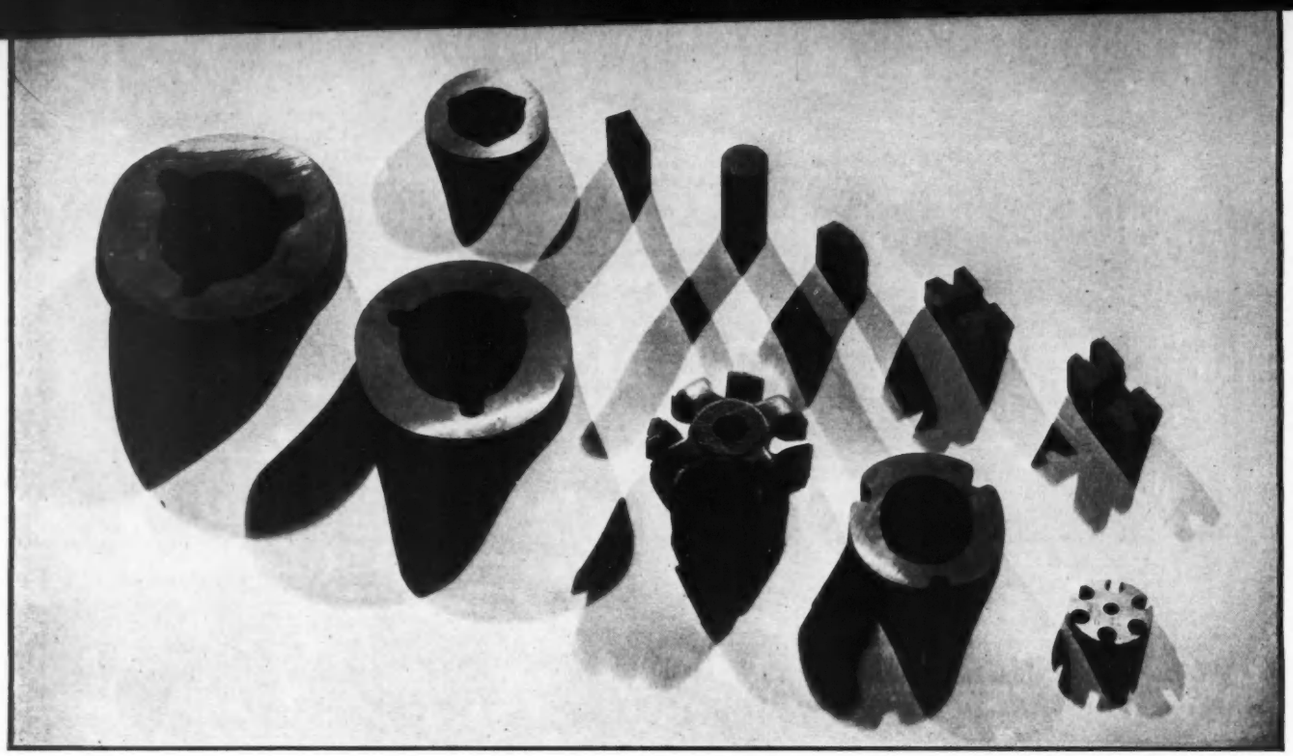
It is consoling of course to those of us who are engineers and research workers connected with industry to be told that we represent the aristocracy of brains of the United States, but compliments of this sort usually involve a certain measure of responsibility. In our case, the responsibility lies in making sure that this aristocracy of ours will be steadily recruited with new brains as the old ones pass away. Where else can we look for such replacement than to our universities and engineering schools?

Now, if we examine the pedigrees of the cerebral aristocrats in the laboratories and engineering departments of industry we find that an appreciable number of them have come from the faculties of these engineering schools that are to train the engineers of tomorrow. Industry can always offer higher salaries and brighter financial prospects than can the universities. Added to this is the inducement of better facilities with which to work. It is not surprising, then, that industry can usually attract the men it needs and wants away from the teaching profession.

But what has been the effect on the university faculties? They are the men who are being relied upon to train the future engineers and research men for industry. The obvious tendency is that, with the few exceptions of those teachers with independent means and those to whom the higher salaries of industry are no particular lure, teachers of lesser capabilities get anchored in the faculties and thus accumulate. It is a most dangerous tendency and one that should be corrected by the industrial companies themselves.

Admittedly in a few instances industry has lost to the university faculties some of its outstanding men who have accepted positions as heads of departments or university deans. This movement from industry to university is desirable, but there is a likelihood that men who take positions of this sort will do more administrative work than teaching.

WHY WOULD it not be a good thing for industry to endow professorships in the leading technical schools and universities, as has been done in a few instances by private endowment, simply to insure that the graduates of those schools which industry will hire, shall be given training under teachers of the highest capabilities? Many industries have endowed scholarships and fellowships, and some of them send their more promising younger employees back to the university for graduate training. Money invested in teachers for these men, and for the undergraduate engineers, would pay returns to industry far greater than it gets on many other things.



Courtesy of Cinaudagraph Corporation

# *Permanent Magnets*

## *I—HOW TO DESIGN THEM*

**New magnet alloys possessing hitherto unattainable magnetic properties, have stimulated great interest in possible applications of magnets in mechanical designs. The author briefly mentions types of magnet materials available and shows how to calculate the dimensions of magnets for specific applications**

J. DELMONTE

THE DISCOVERY and the development of superior permanent magnet materials during the the past few years have made possible new magnetic designs, hitherto considered impractical. Thus, the new magnet materials have also paved the way for improving existing mechanical designs by using permanent magnets in place of mechanical stops and latches and as a substitute for springs. In the analysis of design problems relating to permanent magnets it is imperative not only to apply the correct formulas for the design of the permanent magnets, with a thorough understanding of their significance, but also to choose the correct material and know the limitations imposed by the available methods of fabricating that material.

In the broadest sense, the function of a permanent magnet is to provide a constant source of magnetic energy to an intended application. This type of magnet is distinguished from an electromagnet, which is effective only during the excitation of electrical coils placed around the magnetic core. There are three major classifications covering the applications of permanent magnets:

1. Applications requiring a constant magnetic field: This class applies to electrical indicating instruments, in particular; for example, permanent magnets used in connection with moving coil ammeters, voltmeters, D'Arsonval galvanometers, compass needles, certain relay applications, and miscellaneous uses.

2. Applications utilizing stored magnetic energy: A permanent magnet is said to store magnetic energy, which is manifested when there is a mechanical motion of electrical conductors in the magnetic field. Within the limitations described later, there is no diminution of stored magnetic energy. It is analogous to friction on a surface; an externally applied force is required to show the presence of the friction. Thus, there are in this group vibration pickup devices, phonograph pickup units, brake magnets of watt-hour meters, magnetos, light-weight generators, and similar examples.

3. Applications combining electromagnets and permanent magnets: Some of the earliest applications and some of the latest applications of permanent magnets fall within this classification. The telephone receivers

and loud speakers units are among the earlier applications, while a recently developed line of magnetic switches and relays feature the combination of an electromagnet and a permanent magnet.

The most fertile field for new magnetic designs lies in the second and third classifications. Design possibilities will become apparent later, when certain of the applications are discussed in detail. In some instances, it may prove feasible to replace present electromagnets by permanent magnets.

Permanent magnets were first constructed of hard, high-carbon steel. These were superseded by alloy steels, notably tungsten and chromium steels, which possessed more desirable magnetic properties. An even more satisfactory material was developed by Honda of Japan, in cobalt magnetic steel. However, the cost of cobalt raised the price of these alloys to several times that of the tungsten or chromium steels. As a result, the cobalt steels are used only in special applications in high grade equipment. More recently, alloys of nickel-aluminum steel have stimulated engineering interest in permanent magnets, as they offer properties considerably superior to the earlier types. The cost of these alloys is also comparable. Credit for the development of their magnetic properties is due Professor Mishima of Tokyo. Professor Honda has not been idle, as he has developed a new cobalt-nickel-titanium steel rivaling the properties of Mishima's alloys. The aluminum-nickel magnetic alloys are less expensive than the new Honda steel, and are being produced commercially in this country.

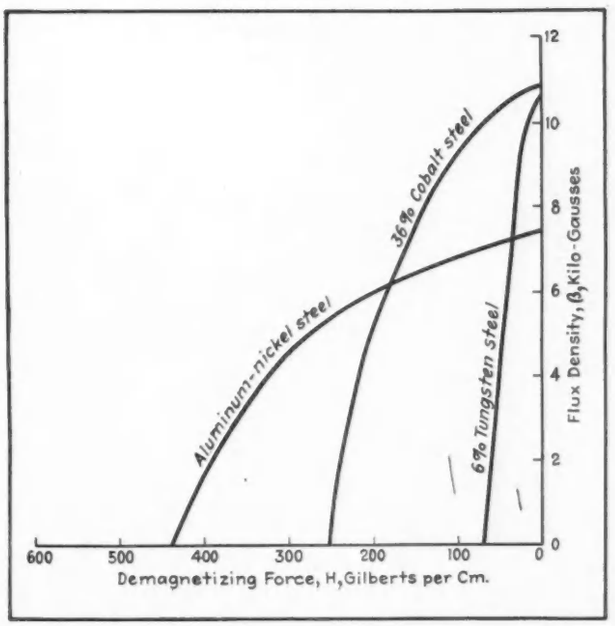



***Fig. 1—Demagnetization curves of some permanent magnet steels, indicating the ability of the material to retain its magnetism against demagnetizing influences***

Of paramount importance to designers are the demagnetization curves, which are shown in Fig. 1 for several types of permanent magnet materials. The ordinates of the demagnetization curves are plotted in gausses, $B$, one gauss being equal to one maxwell or line of magnetic flux per square centimeter. The abscissas are indicated in terms of $H$, the demagnetizing force, which is expressed in gilberts per centimeter, or oersteds. The designers are interested in the available flux density in the material, under the effect of demagnetizing forces. This information is readily obtainable from the demagnetization curves. The significance of the curves will be brought out later.

The methods of determining the flux density required by the intended application are numerous. From an examination of the classifications of permanent magnet applications it will be observed that three fundamental electrical phenomena are involved:

1. Generation of electromotive forces in a coil moving in a magnetic field.
2. Exertion of force upon a conductor carrying current in a magnetic field.
3. Exertion of an attractive force by a magnet upon a magnetic material.

Upon ascertaining the required flux density from the basic formula, designers may proceed with the

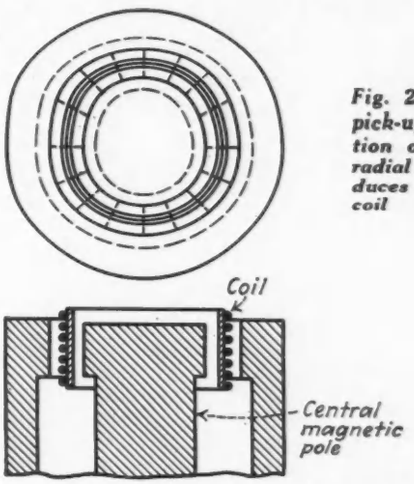



***Fig. 2 — A "vibration pick-up" unit. The vibration of the coil in the radial magnetic field induces a voltage in the coil***

selection of the magnet steel and the calculation of the physical dimensions. A few examples follow herewith, to illustrate the methods of determining flux density.

1. A vibration pickup unit is an example of an application utilizing stored magnetic energy. Fig. 2 illustrates the subject of this example, a coil suspended in a radial magnetic field. Motion of the coil with respect to the poles induces a voltage in the coil, which voltage is highly significant to the measurement, as it is indicative of the velocity of the coil with respect to the magnetic poles. Lenz' law states:

$$e = -N\frac{d\phi}{dt}\,10^{-8}\text{ volts} \qquad (1)$$

where

$e$ = instantaneous value of the electromotive force
$N$ = number of turns in series with the coil
$d\phi/dt$ = rate of change of flux linking the coil

For a circular coil oscillating back and forth in a radial magnetic field, with the motion of conductors perpendicular to the field:

$$e = -B\,N\,d\,v\,10^{-8} \qquad (2)$$

or

$$B = -e\,10^{8}/(N\,d\,v) \qquad (3)$$

where:

$B$ = required flux density in gausses
$d$ = circumference of conductor coil in cm.
$v$ = velocity of conductor in cm. per sec.

The flux density in the gap between the central magnetic pole and the outside pole is determined. With

this information, the design of the magnet may proceed. With the exception of $B$, all of the quantities in Equation (3) are known. The conditions must be studied for arbitrarily determining a value of $v$. The range of the measuring equipment will establish the limits of the voltage, $e$. It usually proves wise to substitute extreme values of $v$ and $e$ in the formula.

2. In the design of some instrument, it is necessary to determine the flux required to exert a certain force on a coil in the magnetic field. This is obtained by the following formula:

$$B = \frac{F\ 10^7}{2.25\ s\ IN} \qquad (4)$$

where

$B$ = required flux density in gausses
$F$ = force in lb.
$I$ = current in coil in amperes
$s$ = length of coil in cm. (perpendicular to the flux)
$N$ = number of turns in coil

In this problem, one must rely upon previous experiences to set arbitrarily the dimensions of the coil and the amount of current passing through the coil. With

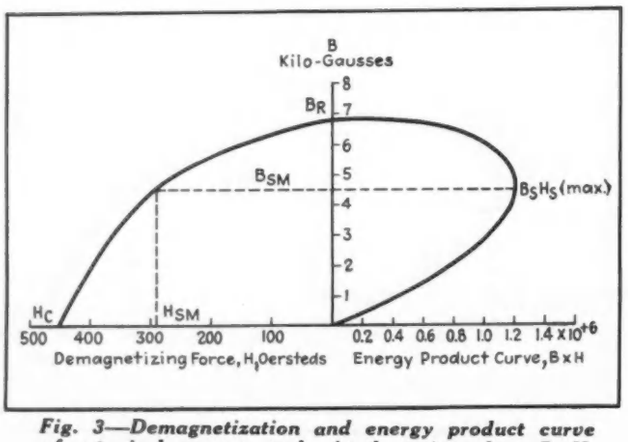



***Fig. 3—Demagnetization and energy product curve of a typical magnet steel. At the point where $B_s$ $H_s$ is a maximum, the volume of magnet steel required is a minimum***

preliminary information of the force the coil must exert, the required flux density is readily calculated.

3. As a further illustration of ascertaining the flux density, the formula for the attractive force exerted by a permanent magnet on a magnetic material is given below. The value of flux density is obtained by referring to a magnetizing curve for the material, in order to ascertain the flux density corresponding to the ampere-turns, $NI$ per centimeter, calculated from the formula below. It is important to remember in this example, that the $NI$ calculated by the formula is not the $NI$ required to magnetize the material but is a term which permits a ready estimate of the flux density required from the final product.

$$NI = L_g c \sqrt{\frac{F_a}{A_s}} \qquad (5)$$

where

$NI$ = ampere turns obtain $B$ (Gausses) from the magnetizing curve of the material at the value of magnetizing force equal to $NI/L$, gilberts per cm.
$L_s$ = length of magnet in cm.
$L_g$ = length of air-gap in cm.
$F_a$ = attractive force in lb.
$A_s$ = cross-sectional area of magnet in sq.cm.
$c$ = constant of the material, depending upon the properties of the magnet, dimensions of the magnet, and degree of saturation. As a trial value, use $c = 4{,}000$.

By the above formulas, or others of a fundamental nature, the flux density required for a particlar application of the permanent magnet may be established. From this point, it is possible to design the magnet efficiently, as to the dimensions of the material and the size and nature of the air-gap. In the final product, it may become necessary to alter slightly the value of flux density to meet unforseen conditions. Likewise, the calculated flux seldom agrees exactly with the observed, though by a careful analysis, it should be quite close. The proportions of the magnets and the air-gaps may be calculated effectively by the methods outlined below.

## *Design of Magnets*

The optimum condition of permanent magnet design occurs when the magnetic energy maintained in the air gap is a maximum per unit volume of magnet steel. At this point, the required flux density is obtained with the least amount of metal. Consider Fig. 3, which illustrates a typical demagnetization curve and an energy product curve for the material. Examine then Fig. 4, a typical bent permanent magnet with an air-gap and two pole shoes. The following terms will be used in the calculations:

$\phi_g = B_g A_g$ = total air-gap flux required from the magnet
$B_g$ = air-gap flux density in gausses, as determined by Equations (3), (4) or (5)
$A_g$ = area of air-gap in sq.cm. (Note: If air gap area is

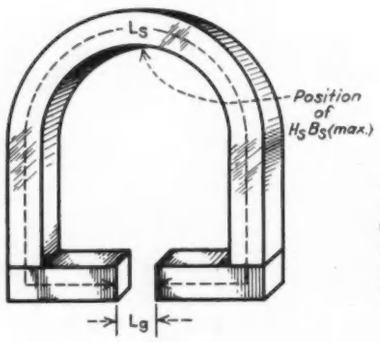



***Fig. 4—Bent permanent magnet wth an air gap and two pole pieces, calculations for which are given in the text***

$a \times b$ or $\pi a^2$, then the effective air gap areas to be used in the calculations are: $(a + L_g)$ $(b + L_g)$ or $\pi\ (a + L_g)^2$
$L_g$ = length of air-gap in cm.
$\phi_s = B_s A_s$ = total flux in magnet steel
$B_s$ = flux density in steel, obtained by initial magnetization
$A_s$ = average cross-sectional area of steel path in sq.cm.
$L_s$ = length of magnetic steel path in cm.
$H_s$ = magnetizing force in gilberts per cm.
$H_{sm}$ = magnetizing force at $H_s$ $B_s$ (max.)
$B_{sm}$ = flux density at $H_s$ $B_s$ (max.)
$\phi_l$ = leakage flux

For theoretical calculations:

$$\phi_g = \phi_s \qquad (6)$$

$$B_g\ A_g = B_s\ A_s \qquad (7)$$

$$A_s = \frac{B_g\ A_g}{B_s}$$

As the total magnetizing force equals magnetic potential across gap:

$$H_s\ L_s = B_g\ L_g$$

and

$$L_s = \frac{B_g\ L_g}{H_s} \qquad (8)$$

$$\text{Volume of magnet steel} = L_s A_s = \frac{B_g^2 A_g L_g}{B_s H_s} \quad (9)$$

When $B_sH_s$ is a maximum, the volume of steel required is a minimum. However, the practical design of the magnet cannot be based upon the ideal economic design. As a result of leakage, the $B_sH_s$ varies from a maximum at the neutral section of the magnet to a minimum at the ends. The highest practical mean value of $B_sH_s$ is regarded as the best commercial practice.

For the practical design of permanent magnets,

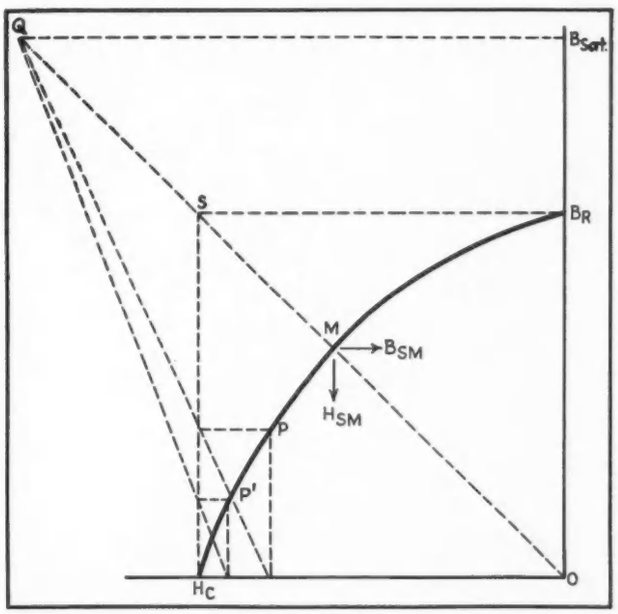



***Fig. 5—Showing how the values of $B_{sm}$ and $H_{sm}$ can be obtained graphically from the demagnetization curve***

leakage factors are introduced, which not only depend upon the size and shape of the magnet, the air-gap and magnetic paths in the vicinity of the magnet, but also depend upon demagnetizing influences of vibration, heat, and adverse fields. Consequently, one finds, for example, that the magnet of an ignition magneto weighs as much as 2 to 3 times that expected for the theoretical economical design, as given by Equation (9). It is possible to apply practical formulas to the design of permanent magnets by modifying Equations (6), (7) and (8) as follows:

$$\phi_s = \phi_1 + \phi_g \quad (10)$$

$$A_s = \frac{B_g A_g f_1}{B_{sm}} \quad (11)$$

where:

$f_1$ = leakage factor = 1.5 to 2.5, which is a practical range. When magnetic paths are close together or large demagnetizing influences exist, high values of $f_1$ should be used

$$L_s = \frac{B_g L_g f_2}{H_{sm}} \quad (12)$$

where:

$f_2$ = leakage factor = 1.1 to 1.5, which is a practical range. For magnet steels with a low $B_s H_s$ (max.), high values of $f_2$ should be used.

The multiplication of typical values of flux density $B$ and demagnetizing force $H$, as obtained from different points on the demagnetizing curve of Fig. 3, gives the energy product values plotted on the right hand side of the graph. At some point $B_sH_s$ (max.) the values $B_{sm}$ and $H_{sm}$ are indicated. These values may be obtained graphically, as indicated in Fig. 5, without the necessity of calculating all of the points for the energy product curve. The graphical method requires information on three characteristics of the permanent magnet material; namely, its coercive force, residual magnetism and saturation flux density.

The coercive force $H_c$ represents a demagnetizing force just necessary to reduce the residual induction to zero, that is, complete demagnetization. High values of coercive force are desirable as they are indicative of the ability of the permanent magnet to withstand demagnetizing influences. It is expressed in terms of gilberts per cm. or oersteds.

Residual magnetism, remanence or retentivity, $B_r$, represents the fraction of the applied magnetizing force retained by the magnet steel after the magnetizing force has been removed. It is expressed in gausses. High retentivity is characteristic of permanent magnet materials.

Saturation flux density, $B_{sat}$, represents the state of magnetism above which it is impossible to induce a further concentration of flux density, by increasing the magetizing force. The material must be magnetized to the saturation flux density in order to retain the maximum possible value of $B_r$ upon the removal of the magnetizing force.

To obtain $B_{sat}$ and $B_sH_s$ (max.) graphically, as in Fig. 5, a horizontal line is drawn through $B_r$, intersecting a vertical line through $H_c$ at $S$. A line from the origin through $S$, intersects an oblique line, plotted from horizontal and vertical lines at an arbitrary point $P$. The intersection of these two lines is at $Q$. Line $O$-$S$ intersects the demagnetization curve at $M$, which point corresponds to $B_sH_s$ (max.). A horizontal line through $Q$ intersects the ordinate axis at $B_{sat}$.

For rapid calculation of $B_sH_s$ (max.) the following formula by Sanford, gives good results:

$$B_s H_s \text{ (max.)} = 0.42\, B_r H_c \quad (13)$$

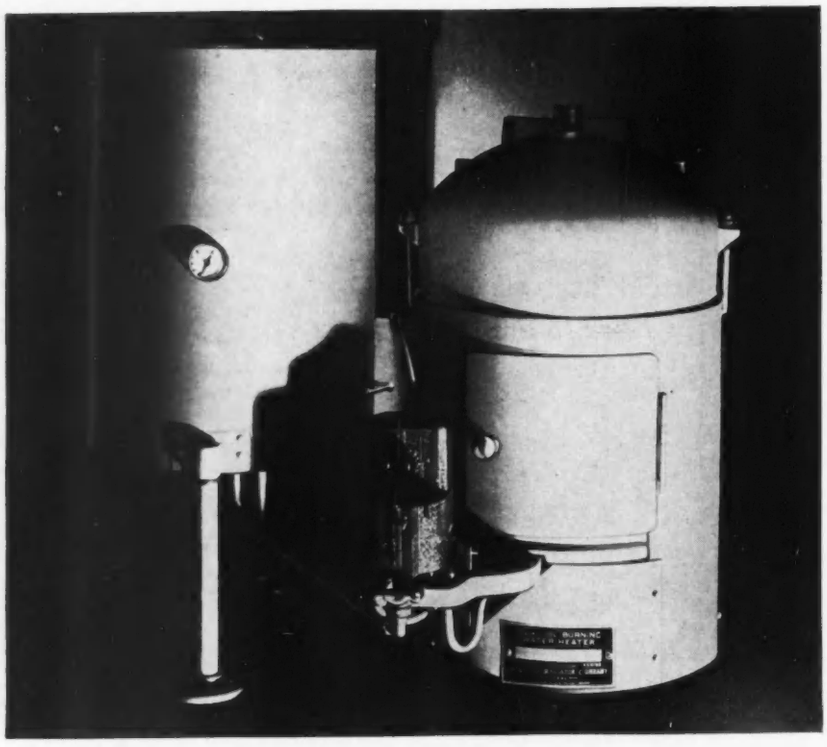
In the comparison of permanent magnet materials, high values of coercive force, residual magnetism, and energy product ($B_sH_s$ max.) are sought. High values of coercive force connote great resistance to demagnetizing influences of heat, vibration, and transient demagnetizing fields.

In reality, the choice of the most desirable permanent magnet material is dependent upon the individual design. Though the natural tendency is to select the materials with the highest $B_s$ $H_s$(max.), these materials cost more. Expensive elements as cobalt, or critical processing and heat-treatments, raise the cost of the more desirable materials. However, the saving in weight and the decreased size may warrant the selection of the newer materials. Smaller volumes of magnet steel are realized, not only as a result of high $B_sH_s$ (max.), but also because, as shown by the practical formulas (11) and (12), they require lower leakage factors.

Having established the magnetic requirements for a specific application, the next step is the selection of the permanent magnet material. In Part II of this article which will appear in May *P.E.*, the author discusses the various permanent magnet materials, gives their properties, processing and typical applications of permanent magnets.

# PRODUCT DEVELOPMENTS

## *Grouping the Units Makes Compact Heater*

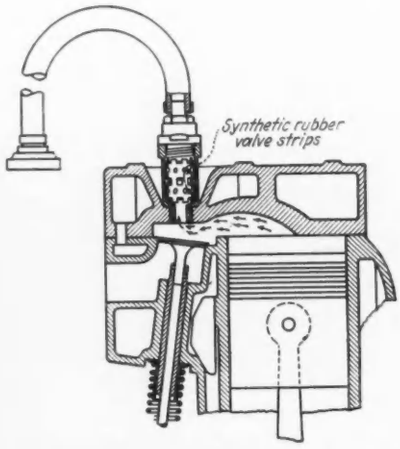


***American Radiator Company's new oil-burning water heater and storage tank unit, designed as a self-contained assembly***

To conserve floor space and to make possible shipment of a completely assembled unit, the American Radiator Company has developed an oil-burning hot water heater for small homes, restaurants, barber shops and similar installations for industrial purposes. This is accomplished by grouping the dome-type heater, gravity oil burner, oil storage tank and controls into one compact unit.

Using a low-priced range oil or distillate, the water heater can be operated either automatically or manually on either of two burners. The small or pilot burner consumes less than $\frac{1}{4}$ lb. of oil per hr., while the larger burner burns 1 lb. of oil per hr. No moving parts are used on the burner or burner shells which are made of non-corrosive stainless steel. The 4-in. pilot and the 6-in. main burner assemblies rest on an iron plate bolted to the heater. The aluminum painted dome type heater has a cast-iron heating section with a steel combusition chamber that is insulated with rock wool. A special steel baffle in the top of the heater keeps the hot gases within the heater for a longer period of time than was done previously.

The 8-gal. oil storage tank, made of 22 gage terne plate, is mounted on adjustable iron legs as shown at the left of the illustration. The strainer at the bottom of the tank is made of brass mesh and copper tubing is used to connect the tank with the heater. A magnetically operated gage is mounted on the outside of the oil storage tank.




***Ability of synthetic rubber to retain resiliency when used as valve strips improves the working efficiency of this tire-inflating device***

## • *Synthetic Rubber Valve-Strips*

Working efficiency has been improved through the use of synthetic rubber for the valve strips of a device known as the Enginair which is inserted in place of a spark plug to inflate motor vehicle tires. Neoprene, the chloroprene rubber manufactured by the du Pont Company, was substituted for the material previously used because the latter became stiff and hard after a relatively short period of operation, and then did not close the intake openings effectively in order to maintain compression.

The new material was found to give the required resiliency and also to withstand the effect of oil from the engine when it happens to come into contact with the strips.

The tire inflater consists of a small valve and hose with a pressure gage and tire-valve connection. The valve, which screws into the spark plug hole, is made up of a cyinder with two rows of perforations drilled around its periphery. Air is drawn through the perforations on the down-stroke of the piston. A cage within the valve cylinder holds the valve strips in the proper position so that on the up-stroke of the piston they close the perforations, and air is forced into the tire.

## • *"Xaloy" Hard Surfacing*

For lining or coating parts where high hardness and resistance to abrasion are desired, but where high impact values and ductility are not important, a material known as Xaloy is announced by the Wilcox-Rich Division of the Eaton Manufacturing Company. The material has a tensile strength of 43,000 lb. per sq. in. and compression strength of 240,000 lb. per sq. in. Its hardness is equivalent to 750 Brinell or 68 to 70 Rockwell C.

Applied in a molten state by a recently perfected method of centrifugal casting, Xaloy is fusion-bonded to the part subjected to abrasive wear. The coating has uniform thickness and hardness. The wall thickness is closely controlled to minimize finishing operations for which special honing and grinding equipment are required. The resulting surface is mirrorlike and has a low coefficient of friction.

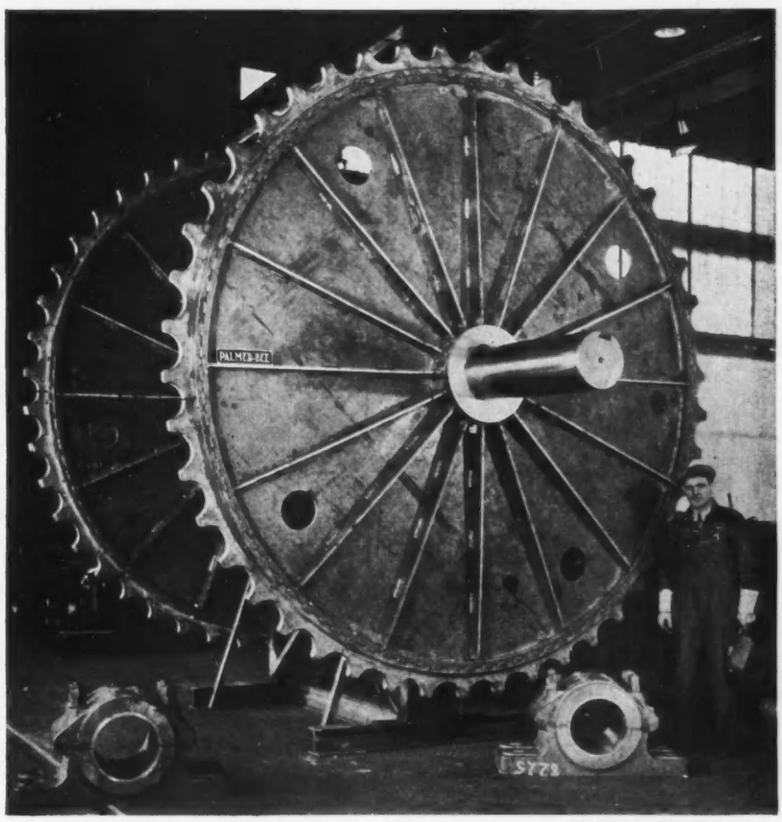
Application of the metal to irregular surfaces is accomplished by means of casting in permanent or sand molds or by the use of inserts faced with Xaloy. Narrow flat surfaces or edges can be coated by the centrifugal-casting method, but sizes are somewhat restricted.

It is reported that the life of tool bushings is lengthened 500 to 600 per cent when lined with Xaloy. Good results have also been obtained by using this alloy as an internal liner for honing control-bushings, the inside diameter of which is subject to a continuous wiping-wear aggravated by the presence of abrasive compound. Another use made of the material has been as a coating on the periphery of rolls used to slit rubber fabrics, against which high-speed steel rotary knives operate. A considerable increase in surface life is reported.

Other application possibilities include cylinder liners, drill and reamer bushings, deep-drawing dies, sizing dies, briquetting molds, ring and plug gages, packing gland sleeves, pilot bars and valves.

## • *Large Sprocket Fabricated by Welding*

Said to be the largest of its type ever made, this 13-ft. sprocket wheel for a vertical core oven was fabricated by electric welding. Whereas formerly it was impossible to maintain outside diameters to within 1 in. of specifications, dimensions of this wheel were held to within $\frac{1}{8}$ in. Manufactured by the Palmer-Bee Company, the wheel consists of a hub 17 in. diameter by 15 in. long with a 12 in. bore, a web $\frac{3}{4}$ in. thick, 12 ft. diameter, stiffened by $\frac{3}{4}$ in. plate braces; and a rim 2 in. thick by 5 in. wide. Eight sectors, steel castings, seven of which have six teeth each and one with four teeth are welded to the rim.




The Lincoln Electric Company

***Large sprocket wheel for vertical core oven fabricated by welding steel castings and steel plates***

## • *Cam Switches for Built-in Use*

Designed for built-in control applications, a new line of rotating cam switches manufactured by the General Electric Company is adaptable to a variety of electrical functions and machine requirements varying from machine motor control to heavy-duty master switching.

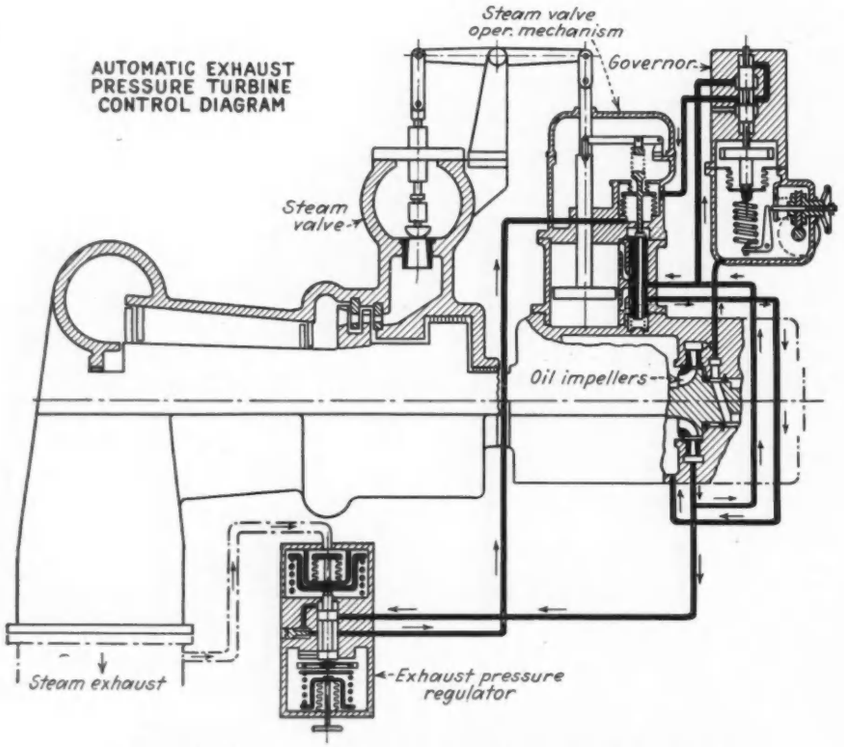
***Rotating cam switch, designed for built-in use, employs flanges for flush-mounting and permits use of protective housing even when inclosed entirely within machine***

The flanges provided for flush-mounting the switches on machines permit them to be used for built-in applications merely by providing an opening and drilling in the face plate of the machine. Since each switch is completely inclosed in a steel housing, even when mounted entirely within a machine, the operator is safeguarded and the possibility of failure due to accumulations of dirt on the switch is minimized. For service where the switches are subjected to a large amount of non-explosive dust, dust-tight covers

having felt gaskets riveted around the edges can be applied. For non-built-in applications the switches are available without the flanges used for flush-mounting.

Chief advantage provided by the adaptability of the new line is that a stock of only a few standard models is said to meet almost any machine requirement. Two sizes are offered: Size 0 rated 15 amp. continuous, and Size 1, rated 25 amp. continuous, both for 600 volts or less a.c. Four standard types are available, as follows, for controlling: (1) single-speed motors for reversing applications; (2) two-speed, separate-winding motors for non-reversing applications, or for one speed forward and one speed reverse; (3) two-speed, single-winding motors for non-reversing applications, or for one speed forward and one speed reverse; and (4) three- or four-speed motors for three or four speeds forward and one speed reverse. In addition special sequences are obtainable and forms of the switches with off-position contacts are available. The switches can also be used as circuit-transfer switches.

## *Hydraulic Governor for Turbine Valve*




***Control diagram of pressure-transformer governor for turbines, hydraulically operated to control the speed automatically and maintain standard generator frequency***

To automatically maintain standard frequency and time, a pressure transformer, which multiplies small fluid pressures to provide sufficient force to move any size valve in the control system for turbines, has been developed by Westinghouse engineers. This new type of hydraulic governor has been embodied into a standard device and is being applied to all turbines of 10,000 kw. and above. It consists of three principal parts; namely, a reverse flow impeller attached to the turbine shaft, a pressure transformer and a servo-motor for operating the governing valves. The mechanism in its simplest form is shown in the accompanying illustration.

The impeller, being mounted on the turbine shaft, delivers a pressure which varies as the square of the turbine speed. Hence changes in this discharge pressure are an accurate indication of changes in rotative speed and provide the governing impulses to the pressure transformer. The transformer multiplies these pressure changes so that the total change in fluid pressure available for operation of the Servo-motor relay is relatively large, and unavoidable friction of relays is of correspondingly little significance. An ample supply of high pressure oil, delivered by the main oil pump, serves as the motive power in the Servo-motor.

Unlike former impeller type governors, there is no actual flow in the discharge from this reverse flow impeller. Therefore, it is said not to be affected by variations in oil demand and the pressure maintained follows speed changes with a high degree of accuracy. One of the features of this control system is that the governor can be made to respond to changes in exhaust pressure, differential pressure or extraction regulation. The regulation can be adjusted from 2 to 8 per cent, and isochronous operation can be obtained by a simple adjustment at the outside of the governor. This arrangement makes it unnecessary for drive arrangements, such as worms for gears, to be connected to the turbine shaft. The speed changer is capable of adjustment over a range of from $\frac{1}{3}$ speed to maximum speed. Time lag can be introduced into the functioning of the governor when desirable.

## • *Random Jottings About New Developments*

Blades of the new line of hand shovels manufactured by the Ames-Baldwin-Wyoming Co., Parkersburg, W. Va., are now being made of stainless steel of No. 2-B finish, as reported by the American Rolling Mill Co. The shovels are designed for use wherever chemicals or chemically treated products are handled, such as for coal and coke that has been sprayed with calcium to minimize dust.

* * *

Spring clips are used in place of screws and nuts by one manufacturer for assembling the two parts of a molded plastic heater plug such as used on some electric irons, toasters and other appliances. A snap ring is slipped over the neck-extensions at the wire entrance and a U-shaped clip holds the contact end together. These fastenings may be removed by a wedge-shaped tool.

# *Selecting Colors for Product Finishes*

## How a typical problem was solved by Gilbert Rohde, and some comments on color selection by Raymond Loewy and Walter Dorwin Teague

GUSTAVE KLINKENSTEIN
*Vice-President and Technical Director*
*Maas & Waldstein Company*

"WE'RE going to finish our new domestic stoker in gunmetal," said the Engineering Department.

"Why gunmetal?" asked the representative of a manufacturer of industrial finishes.

"Our older models were in black or gray, and we want something different for this one. Gunmetal seems about right, so send us samples of your various shades and we'll try them out."

After considerable experimentation and discussion, the engineering department finished a stoker to its own satisfaction and sent the result up to the sales departmen for approval. The sales department looked the new model over and promptly arrived at a unanimous verdict as to its color—"anything but gunmetal!"

As to what color should be adopted, opinion was less harmonious. Finally, however, a fine deep red finish was selected and the job was sent to the production department.

But a few days later the production manager called up the sales manager and informed him that the finish selected could not be used, because that particular finish cost $1.20 more per gallon than the job would stand.

While this particular incident is wholly imaginary, it is a true picture of what happens when the color of a product is considered superficial detail and not as an integral part of the product's design, production, and marketing.

### *How Gilbert Rohde Selects Colors*

By way of contrast, let us consider the color scheme for an oil burner recently designed for the Spencer Heater Company by Gilbert Rohde, the well-known industrial designer. Mr. Rohde regarded color as a major factor in his problem from the very beginning of his work. Tracing back the previous color history of the oil burner, he found that bright colors—reds and greens—had been used for some years because of their appeal to the woman buyer and because they made the product conspicuous in show rooms and exhibits. But a change in color was considered desirable because when other manufacturers adopted the same colors they lost their individuality, and women were asking for colors that would give them more leeway in decorating their basement rooms. Hence it was apparent that the colors to be used should be less striking but no less attractive to women.

Further study developed two other important points. The first of these was that a wrinkle finish, popular among manufacturers because it can be used on rough metal surfaces, was a dust-collector and, therefore, should not be used for this particular purpose. Secondly, the oil burner must look well in the show room, where it is flooded with light, sometimes day-light and sometimes artificial light, but always efficient, and also in the cellar where the light is usually dim. But many colors lose their character and become dull and uninteresting in dim light, so great care had to be taken to select a color combination that would be pleasing under all lighting conditions.

Mr. Rohde's solution of this problem was to use a smooth, glossy lacquer in two shades of bluish-gray for the large areas of the boiler cabinet, with accents of strong red for the burner unit, the valve wheels, and certain decorative touches. Thus, there are sound artistic, functional, and merchandising reasons for every detail of the finish of this oil burner; and a product so designed is well started on the road to commercial success.

### *Some Principles of Color Selection*

Color in product finishing is more important today than it ever was before. The "color movement," which started about 15 years ago, has continued to spread with unabated speed. Products of every description—from tack-hammers to turbines—are becoming more and more colorful. But as soon as the old conventional, safe standards—black, white, and dark gray—are abandoned, the range of choice in colors becomes infinite. How can the designer select intelligently a

color for a fly-swatter, a washing machine, or a portable electric drill?

To Raymond Loewy, the writer is indebted for the following discussion of some of the fundamental principles of color selection.

"Some colors are well liked by most people for their own sake," said Mr. Loewy. "Green, for example, is the favorite color of the American public. Almost any product finished in a green that is not too dark will be well received.

"But what is of more importance from the designer's standpoint, our taste in colors is not static. We like changes in the colors of familiar objects. That is why we are pleased to see green trees turn red or yellow in the fall, and why the first snow-fall is so attractive.

"This desire for color change brings about color cycles in everything we use or wear. In some cases, the cycle is simply from dark to light to dark, but in others the cycle is more complicated—from dark to pastel to light to pastel and back to dark again. As a rule, we tire of light colors more quickly than dark colors, so the dark portion of the cycle tends to be longer than the light; but with some products, the reverse is true. Each individual product has its own cycle, and success in selecting a color that is going to be popular for a given product often depends on the designer's ability to determine in just what phase of its color cycle that particular product happens to be.

"But the rhythm of the color cycle may be upset by causes affecting public taste. Thus, brilliant colors have not been popular for private automobiles since these colors came into general use for taxicabs. On the other hand, there is always an opportunity for some daring innovation, if sound, that will set an entirely new cycle into operation. For example, when we developed the present color scheme for the Greyhound buses, we disregarded all previous treatment of this type of subject.

"One factor that the industrial designer must always bear in mind in selecting colors is practicability. We might like, for example, to furnish a railway club car in pastel shades, but what would be the use? After half a dozen trips the delicacy of the tones would be destroyed for which reason we have to select colors that can withstand service conditions. We have selected the background color of certain floor coverings by taking any kind of carpet and tracking mud over it."

## *Can't Pick Popular Colors in Advance*

That there is no sure rule for selecting in advance new colors that are certain to become popular for wood and metal products, is emphasized by Walter Teague.

"I do not believe it is possible to predict in advance what colors are going to be popular for the coming season," Mr. Teague told the writer. "This, of course, does not hold true for women's wearing apparel where a group of designers and manufacturers are able to control the market and enforce their decisions with reasonable success.

"It is true, however, that in most merchandise there is a slow swing from dark colors to bright ones, and a designer in selecting colors must bear this fact in mind.

"People also are pleased with novel color effects when applied to small things. But radical changes in color applied to large objects, if these are closely associated with people's personal lives, are usually received with hesitancy and doubt.

"As to colors that are preferred for their own sake, most people have personal prejudices in this matter which, however, may change over a period of years. I believe there are no definite rules on the subject, although certain tests have shown that in judging colors, regardless of their application, the majority of women are inclined to favor reds, while the majority of men incline to favor blues."

## *Standard Colors for Mass Production*

As soon as a manufacturer departs from a single "safe" standard color, such as black or dark gray, he has to decide whether he will adopt (1) a new single standard color, or (2) provide a limited selection of colors, or (3) supply any color that any customer may want. The third plan is, of course, out of the question with mass production.

In deciding whether or not it is practical to use a single unconventional standard color, say a light green, one must consider, among other things, the size of the product to be finished. By a law of nature, exemplified by flowers and butterflies, small brightly colored objects will fit into any color scheme, but large ones may not. Thus, if a manufacturer of a can opener finds light green is a popular color, he will be fairly safe in finishing a large number of products in this one color, but not so with a manufacturer of refrigerators. The latter will promptly discover that he can sell light green refrigerators only to women who have, or want, kitchens decorated throughout in light green—and needless to say, his market will be extremely limited. If he is going to offer this color at all, he must also offer others so that the buyer has a rather large selection.

What selection shall be offered? Here are some answers to this question supplied by certain manufacturers of various products.

| | |
|---|---|
| *Kitchen cabinets* | white, ivory (about 50-50) |
| *Refrigerators* | white, (mostly) also cream, ivory pastels, red-and-black |
| *Air Conditioning Units* | light gray (for basement installations), ivory, a brown that is half way between walnut and mahogany, grained mahogany. |
| *Oil heaters* | red, green, gray, light tan. |
| *Store scales* | red, white, bronze |
| *Cash registers* | mahogany, green-black wrinkle finish. |
| *Typewriters* | black japan, black crackle varnish over dark green. |
| *Heavy machinery* | black, steel gray, deep green, deep maroon. |

## *Color Selection a Job for a Specialist*

There can, of course, be no final answer to this question of selecting color standards. The one certain thing is that the spirit of the times calls for variety and change.

The practical solution of the problem, in cases where considerable expense is justified, is to place the responsibility for the selection of color and finish on some one who is thoroughly versed in industrial design, who can study the subject of color selection exhaustively in all its phases, and who will have good reasons for any color scheme that he may recommend for a product.

# *Torque Motors and Their Applications*

**Because torque motors can remain stalled across the line and continue to exert full torque indefinitely, they are used extensively to actuate pulling, clamping or holding mechanisms**

F. W. JESSOP
*President, The Ohio Electric Mfg. Company*

DESCRIBED in simplest terms, a torque motor is a magnet or solenoid having a core element that rotates instead of one that moves axially. In construction it is a high torque motor capable of 100 per cent slip. Thus, the special characteristic of the torque motor is its ability to drive to a stalled position, and, with current on, continue to exert a turning moment for an indefinitely long period of time without damaging the winding through overheating.

Torque motors usually are specially made polyphase squirrel-cage motors. They are designed so that the slip between the rotor inductors and the rotating field is high. Consequently the frequency of the induced emf. is high, and under stalled conditions equals the impressed emf. frequency. The impedance of the rotor inductors is dependent upon the frequency of the induced current and will be at a maximum when the rotor is not revolving.

High resistance is built into the rotor circuit to enable the motor to operate at 100 per cent slip. The operation of the motor compares with a shunt-wound direct-current motor operating with resistance in series with an armature winding.

Slow synchronous speeds are generally desirable for picking up the load easily. To obtain this condition the stator is wound to produce the maximum number of poles for the given number of slots, using more than the usual amount of copper to keep the heat rise at a low value.

Most torque motors are wound for 2 or 3 phase a.c. service up to 550 volts. However, to a limited extent, companion motors are made for d.c. service up to 250 volts, and also for single phase operation.

The frame size of the motor depends upon the magnitude of the torque required to perform the operation, and also upon the time cycle of the working and the idle periods. Thus on a basis of equal torques, a motor which operates six out of sixty minutes uses a frame much smaller in size than a motor that is designed to hold a load for several hours.

Torque motors are usually designed to suit the requirements of a particular application. Factors which enter into the design are; the method of mounting, duty cycle, magnitude of torque, and the nature of the work to be done. As to operation they are made for continuous or occasional rotation, to reverse under load or to drive in one direction only, to remain stalled for hours, and also to operate at high speeds and at low speeds during different parts of a cycle.

Typical standard duty specifications providing for various cycles of operation are:

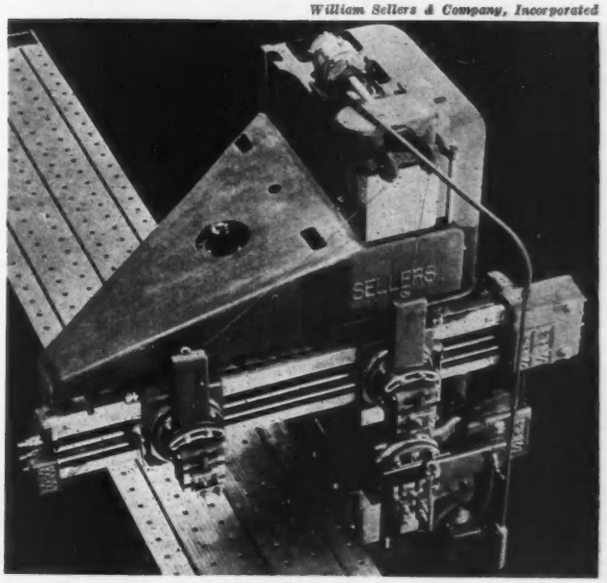
For service where the motor may remain locked across the line for several hours, a "100 per cent locked service type" is used, which can be left locked across the line continuously without injury to the motor.

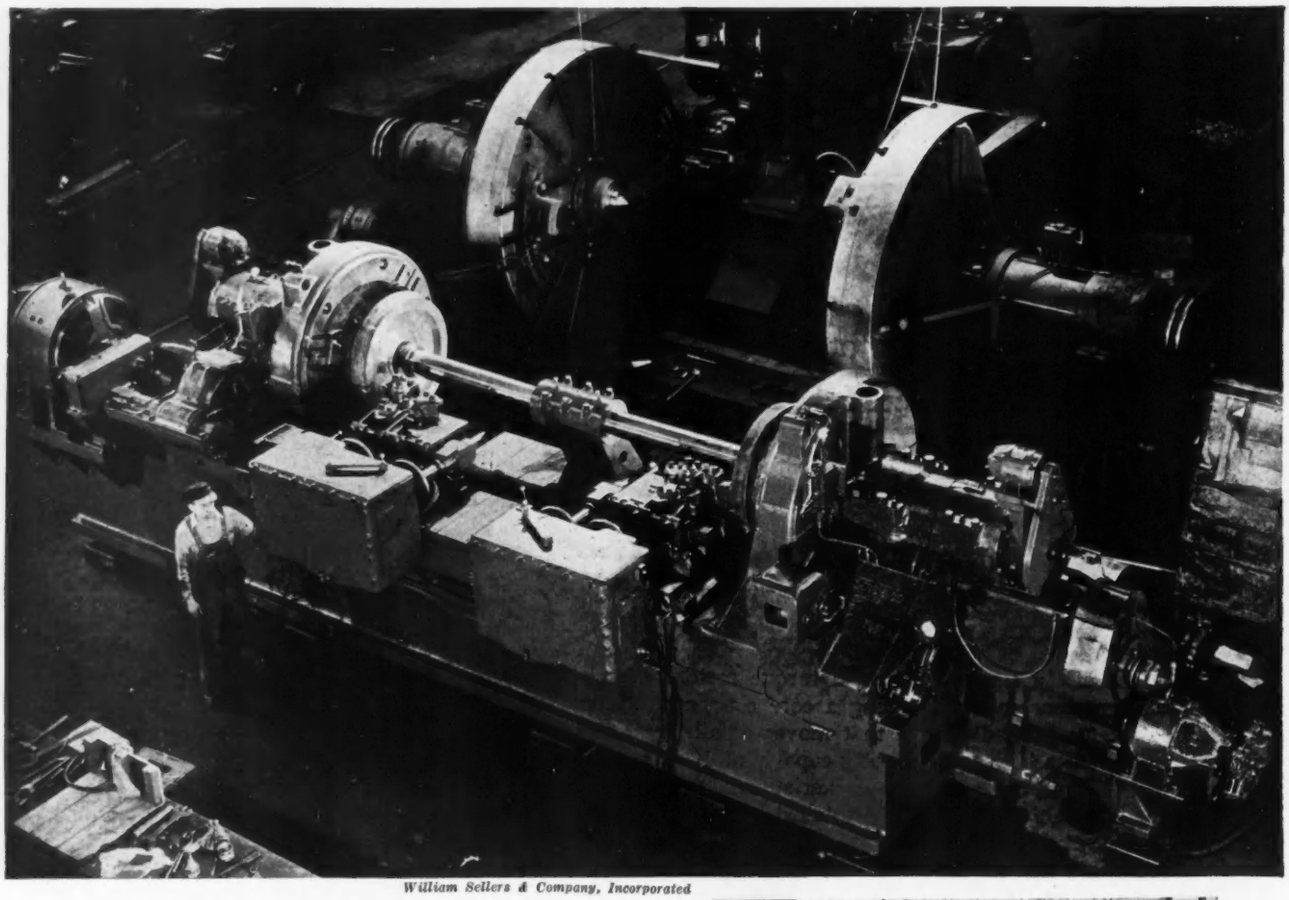
For services where the motor is locked across the line approximately 50 per cent of the time, as for example, approximately 30 seconds on and 30 seconds off, a "50 per cent locked service type" is used, which can be left locked across the line for a maximum time of one hour continuously without injury to the motor.

For services where the motor is locked across the line approximately 25 per cent of the time, as for example,

***Fig. 1—The vertical motor, top end of which can be seen in the top of cross rail brace on this 60 in. open-side planer, is used for clamping the rail***

*William Sellers & Company, Incorporated*





*William Sellers & Company, Incorporated*

***Fig. 2—The small motors located on top of main spindle bearing caps are torque motors for moving spindles in and out when chucking***

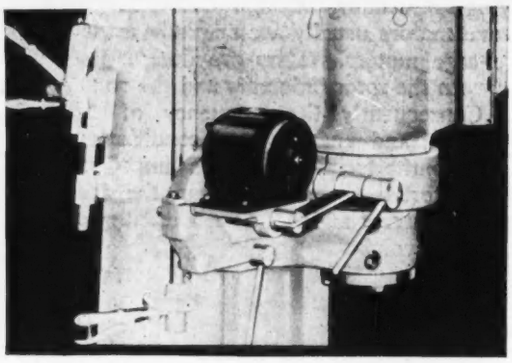


***Fig. 3—Torque motor used to operate clamps that lock the arm of the radial drill to column***

approximately 15 seconds on and 45 seconds off, the "25 per cent locked service type" is used, which can be left locked across the line for a maximum time of 20 minutes continuously without injury to the motor.

For services where the motor is locked across the line approximately 10 per cent of the time, for example, approximately 6 seconds on and 54 seconds off, the "10 per cent locked service type" is used, which can be left locked across the line for a maximum time of 10 minutes without injury to the motor.

These standards can often be altered to fit special conditions.

When the motor is arranged to drive in only one direction, weights or springs in connection with linkage are used to return the parts to the original position for a new start.

Torque motors are almost invariably equipped with ball bearings to insure free operation. They can be made with open, splash-proof or closed frames.

Torque motors are wound so that the full rated torque is developed while the rotor is locked or stalled in any position, or while rotating slowly. At no-load the speeds are nearly synchronous and the r.p.m. depends upon the number of poles and the frequency of the line current. Slow speeds are generally desirable so that the motion may be accomplished with little shock, noise or vibration.

The pull of the motor is uniform. The current required is small and does not change much during operation. A torque motor with its characteristic impedance requires only from 10 to 20 per cent of the volt-ampere inrush when starting or placing on the line as that needed by a solenoid of the same rating with an open magnetic circuit.

For driving mechanisms such as pushing, pulling and holding devices, the torque motor has found many applications. Formerly used almost exclusively in connection with elevating and hoisting machinery, these motors are now being installed on many other types of machines. A pinion, sprocket, or sheave wheel keyed to the motor shaft is used in many arrangements to drive gear trains, chain wheels, or lifting devices which are connected to the machine part that is moved and held. They are used to perform the same types of operations as are ordinarily done by electric solenoids, friction drives, air or hydraulic cylinders.

An interesting application of a torque motor used for clamping the rail of a 60 in. open-side planer is shown in Fig. 1. Mounted on the top of the movable cross

rail brace, the motor is controlled from a pushbutton station at the end of the cross rail where it is easily reached by the operator. The motor has 144 lb. in. torque capacity at a five minute rating.

The 42 in. car wheel lathe for contouring shown in Fig. 2 is equipped with two torque motors of 240 lb. in. torque having a five minute, 50 deg. C. continuous rating. These motors are mounted on top of the machine and are used for moving the spindles in and out when chucking wheels. It is a production type of machine and the motors are used about six times an hour.

Smaller torque motors have been used on feed change boxes of planer-type millers for throwing the clutches, and on double cut planers to move the tool into position at each reversal of the table.

For clamping the arm of a radial drill to the column as shown in Fig. 3, a reversing type torque motor is used. The motor pinion drives a gear which has a threaded center bushing nut which turns on the threaded section of a plunger in the locking system. The gear is constrained by the housing so that it cannot move endwise. As the gear turns the plunger moves endwise and locks or unlocks the clamp depending upon the direction of rotation of the motor.

An example taken from the elevating machinery field is shown in Fig. 4. Here a torque motor mounted on the machine frame is used to release a hoist brake. A pinion on the motor shaft engages a quadrant arm connected to a toggle brake shoe clamping arrangement, so that when the torque motor is energized the quadrant arm is rotated through an angle of about 60 deg. to release the brake shoes against a spring resistance. When power is cut off the spring returns the quadrant and motor to the original position, setting the brakes.

To permit an elevator gate to be opened, the torque motor shown in Fig. 5 retires a cam by means of a linkage motion. The cam is mounted on the elevator car for operating the door-interlocks at the various landings. The motion of retiring the cam releases the door interlock, which in turn locks the door when closed and completes the control circuit. The elevator car can then be started. The cam remains in the retired position while the elevator car is in motion, and passes the door interlocks of all the intermediate landings without striking.

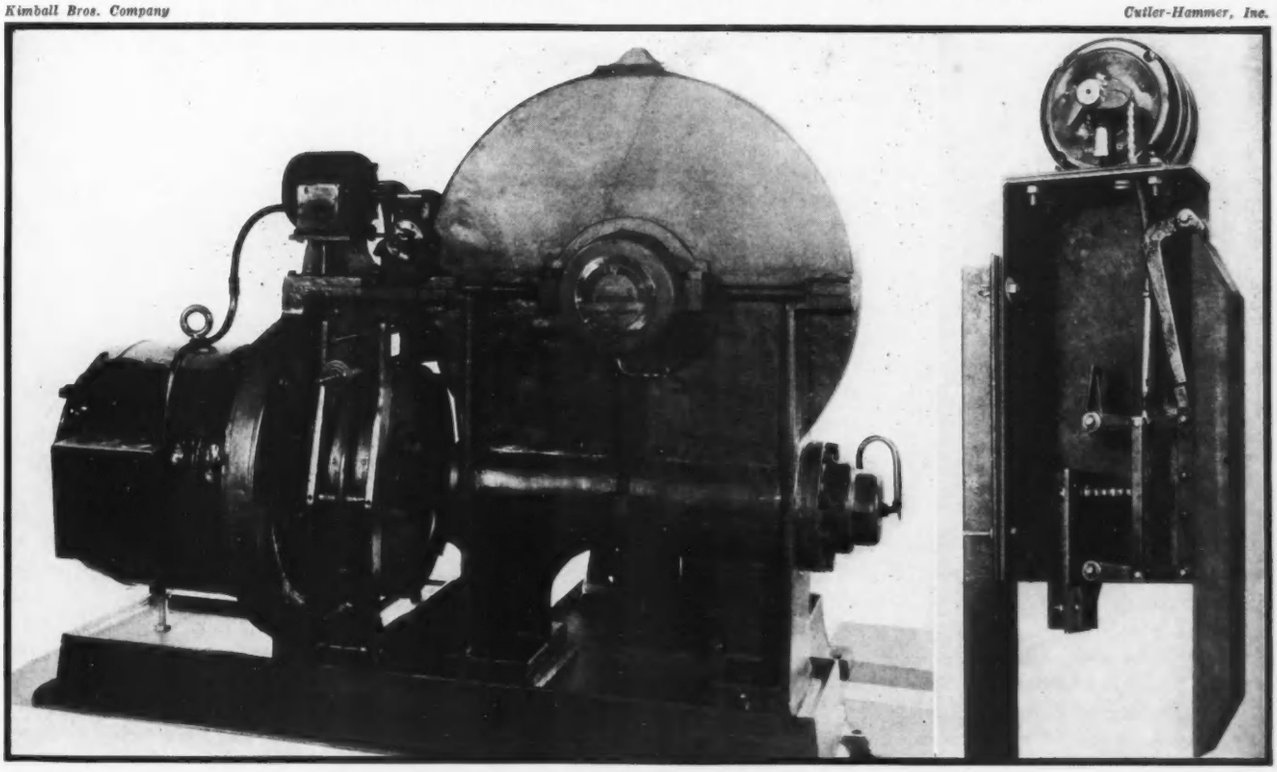
The hardwood cam is mounted on a folding lever mechanism which is operated by a torque motor designed for 50 per cent locked service. With the car in motion the motor is energized and holds the cam in the retired position until just before the desired landing is reached.

Fig. 5 shows the cam in the locking position as it is when the car is stopped at the selected landing with the torque motor deenergized and after the compression springs have extended the linkage system. When restarting the car, depressing the landing button energizes the torque motor which then rotates and moves the cam inwards horizontally by lifting the center link of the mechanism. This is done through the chain connection from the eccentric link on the rotor shaft to the center link of the mechanism. After the limited inward motion of the cam is halted by the mechanical stops the motor stalls and continues to exert a torque until deenergized. When current to the motor is cut off the compression springs again extend the cam and return the motor to its idle position.

Torque motors are also used to open and close lathe power chucks, holding or clamping fixtures and valves, also to operate draft controls, timing devices, tension controls, machine feeding arrangements and locking devices, as well as elevator doors and control equipment.

***Fig. 4—(Left) Mounted on frame over brake, a torque motor drives gear sector to release spring-loaded brake. Fig. 5—(Right) Retiring cam mechanism for elevator service is operated by torque motor and compression springs***

*Kimball Bros. Company* *Cutler-Hammer, Inc.*



# Formulas for Torsional

IN THE table which follows are given formulas for calculating the angle of twist and the shear stress produced in elastic bars and tubes by torsion. The formulas given are based upon the following assumptions:

(1) The bar is straight, has a uniform cross-section, and is of homogeneous isotropic material.

(2) The bar is loaded only by equal and opposite twisting couples, which are applied at its ends in planes normal to its axis. This implies that the end sections are free to warp, there being no constraining forces that tend to hold them in a plane.

(3) The bar is not stressed above the elastic limit.

The notation used and the units in which the several quantities are expressed are given in the table. It should be noted that all lengths are to be expressed in inches, all areas in square inches, all moments of inertia in inches-fourth, and all forces in pounds.

## *Basis and Accuracy of Formulas*

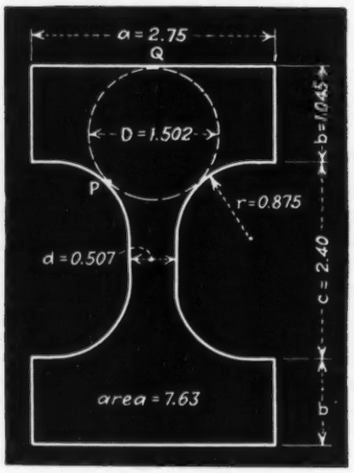
With one exception, for Cases 1 to 7 inclusive the formulas for twist and stress are based on rigorous mathematical analysis, and may be regarded as exact. The original solution was developed by St. Venant. (See Todhunter and Pearson's "History of the Theory of Elasticity," and also the other references given in this article.) The exception is Case 4, the equations for which are given in a simplified form involving an approximation, with a resulting error not greater than 4 per cent. The $K$ formulas for Cases 8 to 16 inclusive and the stress formulas for Cases 8 to 13 inclusive are based on analysis but are approximate. (See J. Prescott's "Applied Elasticity.") Their accuracy depends upon how nearly the actual section conforms to the assumptions indicated as to form. The $K$ formulas for the remaining cases, and the stress formulas for Cases 14 and 20 inclusive, are based on the "membrane analogy," and are to be regarded as reasonably close approximations, giving results that are rarely as much as 10 per cent in error.

[See (U.S.) National Advisory Committee for Aeronautics, *Report No. 334,* "The Torsion of Members Having Sections Common in Aircraft Construction," by G. W. Trayer and H. W. March. See also (British) Advisory Committee for Aeronautics, *Reports and Memoranda, No. 333,* "The Use of Soap Films in Solving Torsion Problems," by G. I. Taylor and A. A. Griffith.]

## *The Soap Film Analogy*

The soap-film membrane analogy is of such great value in guiding judgment as to the torsional strength and stiffness of various solid sections that a brief statement of it will be given here. Essentially, it amounts to this: If in a thin flat plate holes are cut having the outline of various sections, and over each of these holes a soap film is stretched and slightly distended by a pressure from below, the volumes of the bubbles thus formed are proportional to the torsional rigidities of the corresponding sections, and the slope of the bubble membrane at any point is proportional to the shear stress caused at that point of the corresponding section by a given twist per unit length of bar.

An understanding of this analogy makes it possible to draw certain conclusions as to the comparative torsional properties of different sections by simply visualizing the bubbles that would be formed over holes of corresponding size and shape. From the volume relationship, it can be seen that of two sections having the same area, the one more nearly circular is the stiffer, and that while any extension whatever of the section increases its torsional stiffness, narrow outstanding flanges and similar protrusions have little effect. It is also apparent that any thin flat plate, when bent into the form of an open tube or into a channel or angle section, has practically the same torsional stiffness as in its original form. From the slope relationship it can be seen that the greatest stresses (slopes) in a given section occur at the boundary adjacent to the thicker portions, that the stresses are very low at the ends of narrow outstanding flanges or protruding corners, and very high at points where the boundary is concave and at the corners of reentrant angles. A longitudinal slot or groove, if sharp at the bottom or if narrow, will therefore cause high local stresses and if such a groove is deep it will greatly reduce the torsional stiffness of the sections through which it runs.




It must be remembered that all stress formulas give elastic stress, and if used to predict the twisting moment that will cause rupture, will give a result too low. This is particularly true of sections in which high local stresses occur. Torsion tests on specimens made of plaster showed that the breaking torques calculated by the stress formulas were nearly correct for specimens of circular, square and nearly square sections, but gave results about 30 per cent too small for long rectangles and for I's and T's of large fillet radius, while for I's of very small fillet radius the actual breaking torques were as much as eight times the calculated values. This is because the local stresses are greatly reduced by plastic yielding when the elastic limit is passed, even in relatively brittle materials.

The membrane analogy as stated, and the above generalizations based upon it, hold only for solid sections.

# Stress and Deflection

RAYMOND J. ROARK
*Professor of Mechanics,*
*University of Wisconsin*

But it is possible to make somewhat similar generalizations concerning tubular or hollow sections from the formulas given for Cases 7 to 11 inclusive. These show that the strength and stiffness of a hollow section depend largely upon the area inclosed by the median boundary. For this reason a circular tube is stiffer and stronger than one of any other form, and the more nearly the form of any hollow section approaches the circular, the greater will be its strength and stiffness. It is also apparent from the formulas for stress that even a local reduction in the thickness of the wall of a tube, such as would be caused by a longitudinal groove or slot, may greatly reduce its strength, though if the groove is narrow the effect on stiffness is small.

## *Effect on End Constraint*

For any except a circular bar, twisting is accompanied by a warping of the cross sections, and if this warping is prevented, as might be done by welding or gluing the ends of the bar to rigid blocks or plates, the torsional stiffness is increased. In such structural members as I-beams and channels the increase may be great, especially if the member is short. It is not possible to express this stiffening effect by any general formula, but for channels, I-beams and Z-bars the result of complete constraint at both ends is to increase $K$ by an amount approximately equal to

$$\Delta K = \frac{6 E I d^2}{G l^2}$$

where $E$ represents the modulus of elasticity in tension and compression, $I$ represents the moment of inertia of the section about a central axis parallel to the web, $d$ is the depth of the section center to center of flanges and $l$ is the length between the constrained ends. For constraint at one end only, $\Delta K$ has one-fourth of the above value. It is here assumed that the total angle of twist $\theta$ is small.

For bars of simple cross-section, the use of the formulas in calculating angular deformation and stress offers no difficulty. For the more complex sections, such as the I, T or L, the computations are somewhat less simple, and will be illustrated by a typical example.

The section illustrated is that of an airplane wing spar made of spruce, for which $E$ may be taken as 1,500,000 lb. per sq.in. and $G$ as 100,000 lb. per sq.in. It is required to find the angle through which a spar of this form 8 ft. long would be twisted by end couples of 500 in. lb., and to determine the maximum resulting stress.

## *Solution*

All relevant dimensions of the section are shown on the sketch, with notations corresponding to those used in the formulas. The first step is to compute $K$ by the formula given for Case 17, and we have:

$$K = 2K_1 + K_2 + 2\alpha D^4$$

$$K_1 = ab^3\left[1/3 - 0.21\,(b/a)\left(1 - \frac{b^4}{12\,a^4}\right)\right]$$

$$= 2.75 \times 1.045^3\left[(1/3) - \left(\frac{0.21 \times 1.045}{2.75}\right)\left(1 - \frac{1.045^4}{12 \times 2.75^4}\right)\right]$$

$$= 0.796$$

$$K_2 = (1/3)\,cd^3 = (1/3)(2.40)(0.507^3) = 0.104$$

$$\alpha = \frac{0.507}{1.045}\left(0.150 + \frac{0.1 \times 0.875}{1.045}\right) = 0.1133$$

$$K = 2\,(0.796) + 0.104 + 2 \times 0.1133\,(1.502)^4 = 2.85$$

Therefore

$$\theta = \frac{Tl}{KG} = \frac{(500)(96)}{2.85(100{,}000)}$$

$$= 0.168 \text{ radians} = 9.64 \text{ deg.}$$

The maximum stress will probably be at $P$, the point where the largest inscribed circle touches the section boundary at a fillet. The appropriate formula is:

$$S = \frac{T}{K} C$$

$$C = \frac{D}{1 + \pi^2 D^4/(16A^2)}\,(F)$$

$$F = 1 + \left\{0.118 \log\,[1 - (D/2r)] - 0.238\,D/(2r)\right\} \tan \text{h}\,(2\phi/\pi)$$

$$F = 1 + \left[0.118 \log\left\{1 - 1.502/2\,(-0.875)\right\} - 0.238\frac{1.502}{2\,(-0.875}\right] \text{hyp. tan}\,\frac{2\left(\frac{\pi}{2}\right)}{\pi}$$

Substituting the proper values of $T$, $K$ and $C$ it is found that $S = 303$ lb. per sq.in.

It will be of interest to compare with this the stress at $Q$, the other point of contact between the inscribed circle and the boundary, and here the formula that applies is:

$$S = \frac{T}{K} C$$

$$C = \frac{D}{1 + \frac{\pi^2 D^4}{16A^2}}\left[1 + 0.15\left(\frac{\pi^2 D^4}{16A^2} - \frac{D}{2r}\right)\right]$$

Because $r$ is infinity, $D/2r = 0$

By substitution,

$$C = 1.437$$

$$S = 252 \text{ lb. per sq. in.}$$

If the ends of the spar are constrained, as by gluing in filler blocks, the stiffness will be increased, the increase in $K$ being

$$\Delta K = \frac{6\,EI\,d^2}{Gl^2}$$

$$= \frac{6\,(1{,}500{,}000)\,(3.78)\,(3.445)^2}{(100{,}000)\,(96)^2} = 0.44$$

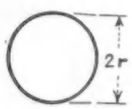
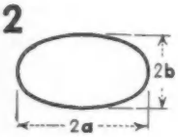
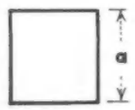
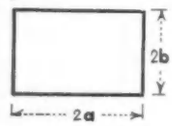
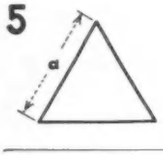
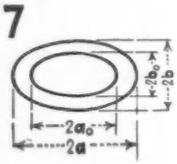
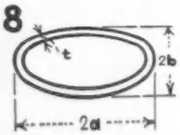
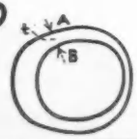
Hence $\theta$ becomes: $\frac{2.85\,(9.64)}{2.85 + 0.44} = 8.35$ deg.

The effect of end constraint would be much less in the case of a metal beam, because the ratio of $E$ to $G$ would be only about 2.5 instead of 15, as for spruce.

(R. J. ROARK)

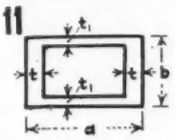
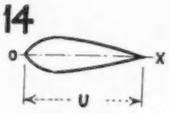
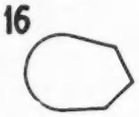
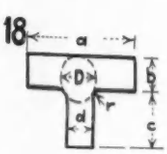
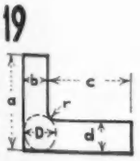
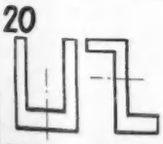
# FORMULAS FOR TORSIONAL

General formulas: $\Theta = TL/KG$, $S = T/Q$, where $\Theta$ = angle of twist (radians); $T$ = twisting moment (in.-lb.); $L$ = length (inches);

| Form and Dimensions of Cross-sections, other Quantities involved, and Reference Number | Formula for $K$ in $\Theta = TL/KG$ | Formula for shear stress |
|---|---|---|
| **1**   Solid circular section. | $K = \frac{1}{2}\pi r^4$ | Max. $S = \frac{2T}{\pi r^3}$ at boundary |
| **2**   Solid elliptical section. | $K = \frac{\pi a^3 b^3}{a^2+b^2}$ | Max. $S = \frac{2T}{\pi ab^2}$ at ends of minor axis |
| **3**   Solid square section. | $K = 0.1406\,a^4$ | Max. $S = \frac{T}{0.208a^3}$ at mid-point of each side |
| **4**   Solid rectangular section. | $K = ab^3\left[\frac{16}{3} - 3.36\frac{b}{a}\left(1 - \frac{b^4}{12\,a^4}\right)\right]$ | Max. $S = \frac{T(3a+1.8b)}{8a^2b^2}$ at mid-point of each longer side |
| **5**   Solid triangular section (equilateral). | $K = \frac{a^4\sqrt{3}}{80}$ | Max. $S = \frac{20T}{a^3}$ at mid-point of each side |
| **6** Hollow concentric circular section. | $K = \frac{1}{2}\pi(r_1^4 - r_o^4)$ | Max. $S = \frac{2Tr_1}{\pi(r_1^4 - r_o^4)}$ at outer boundary |
| **7**   Hollow elliptical section, outer and inner boundaries similar ellipses. $q = \frac{a-a_o}{a_o} = \frac{b-b_o}{b_o}$ | $K = \frac{\pi a_o^3 b_o^3}{a_o^2+b_o^2}\left[(1+q)^4 - 1\right]$ | Max. $S = \frac{2T}{\pi a_o b_o^2[(1+q)^4-1]}$ at ends of minor axis on outer surface |
| **8**   Hollow, thin walled elliptical section of uniform thickness. $U$ = length of median boundary, shown dotted. $U = \pi(a+b-t)\left[1+0.27\frac{(a-b)^2}{(a+b)^2}\right]$, approx. | $K = \frac{4\pi^2 t\,[(a-\frac{1}{2}t)^2\,(b-\frac{1}{2}t)^2]}{U}$ | Average $S = \frac{T}{2\pi t\,(a-\frac{1}{2}t)\,(b-\frac{1}{2}t)}$ (Stress nearly uniform if $t$ is small) |
| **9** Any thin tube of uniform thickness. $U$ = length of median boundary, $A$ = mean of areas enclosed by outer and inner boundaries, or (approx.) area within median boundary. | $K = \frac{4A^2t}{U}$ | Average $S = \frac{T}{2tA}$ (stress nearly uniform if $t$ is small) |
| **10**   Any thin tube. $U$ and $A$ as for Case 9; $t$ = thickness at any point. | $K = \frac{4A^2}{\int \frac{du}{t}}$ | Average $S$ on any thickness $AB = \frac{T}{2tA}$ (Max. $S$ where $t$ is a minimum) |

# DEFORMATION AND STRESS

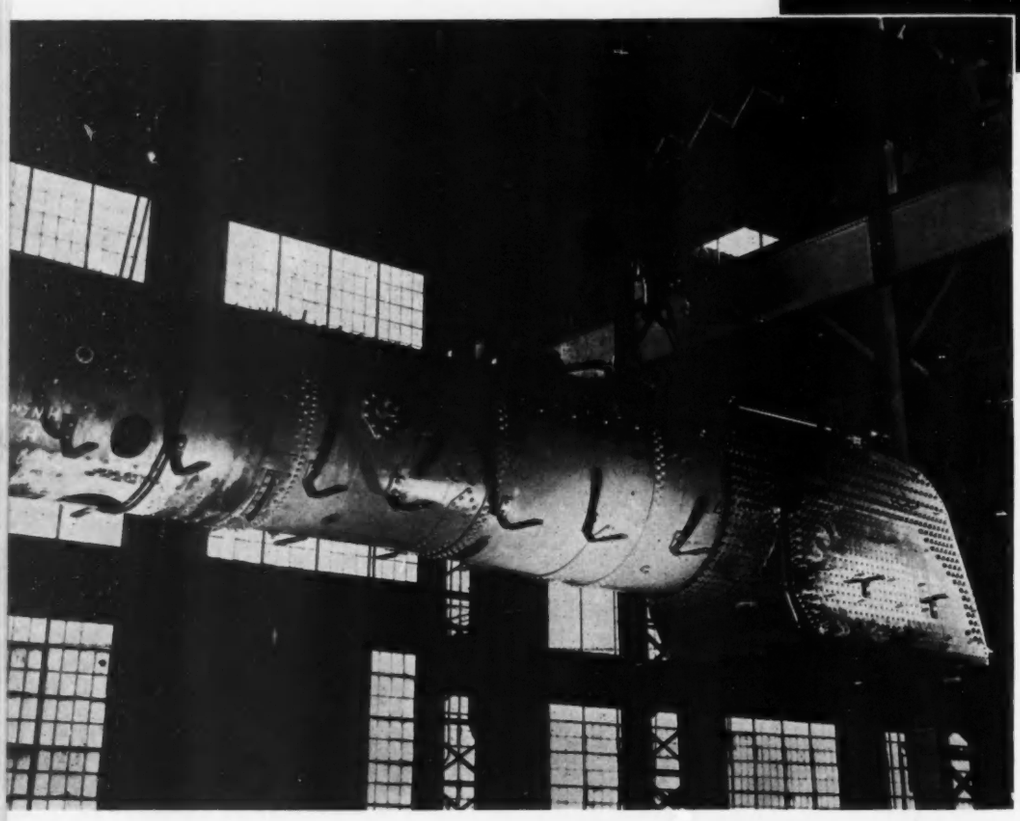
$S$ = unit shear stress lb. per sq. in.; $G$ = modulus of rigidity (lb. per sq. in.); $K$ (in.⁴) and $Q$ (in.³) are functions of the cross-section.

| Form and dimensions of Cross-sections, other Quantities involved, and Reference Number | Formula for $K$ in $\theta = TL/KG$ | Formula for shear stress |
|---|---|---|
| **11**  Hollow rectangle. | $K = \dfrac{2\,t\,t_1\,(a-t)^2\,(b-t)^2}{at+bt_1-t^2-t_1^2}$ | Average $S = \dfrac{T}{2t(a-t)\,(b-t_1)}$ near mid-length of short sides<br>Average $S = \dfrac{T}{2t_1(a-t)(b-t_1)}$ near mid-length of long sides<br>(There will be higher stresses at inner corners unless fillets of fairly large radius are provided.) |
| **12** Thin circular open tube of uniform thickness.<br>$r$ = mean radius. | $K = \dfrac{2}{3}\,\pi\,r\,t^3$ | Max. $S = \dfrac{T(6\pi r+1.8t)}{4\,\pi^2 r^2 t^2}$, along both edges remote from ends.<br>(This assumes $t$ small compared with mean radius; otherwise use formulas given for Cases 14–20.) |
| **13** Any thin open tube of uniform thickness.<br>$U$ = length of median line, shown dotted. | $K = \dfrac{1}{3}\,U\,t^3$ | Max. $S = \dfrac{T(3\,U+1.8t)}{U^2 t^2}$, along both edges remote from ends.<br>(This assumes $t$ small compared with least radius of curvature of median line; otherwise use formulas given for Cases 14–20.) |
| **14**  Any elongated section with axis of symmetry $O\,X$. $U$ = length, $A$ = area of section, $I_x$ = moment of inertia about axis of symmetry. | $K = \dfrac{4\,I_x}{\left(1+\dfrac{16}{3}\,\dfrac{I_x}{A\,U^2}\right)}$ | For all solid sections of irregular form (Cases 14–20 inclusive) the max. shear stress occurs at or very near one of the points where the largest inscribed circle touches the boundary, unless at some other point on boundary there is a sharp reentrant angle, causing high local stress, and of these, at the one where the curvature of the boundary is algebraically least. (Convexity represents positive, concavity negative, curvature of the boundary.) At a point where the curvature is positive (boundary of section straight or convex) this max. stress is given approximately by: |
| **15** Any elongated section or thin open tube.<br>$d\,U$ = elementary length along median line, $t$ = thickness normal to median line, $A$ = area of section. | $K = \dfrac{\frac{1}{3}\,F}{\left(1+\dfrac{4}{3}\,\dfrac{F}{A\,U^2}\right)}$ where $F = \int_0^u t^3\,du$ | |
| **16**  Any solid, fairly compact section without reentrant angles.<br>$J$ = polar moment of inertia about centroidal axis; $A$ = area of section. | $K = \dfrac{A^4}{40\,J}$ | $S = G\dfrac{\theta}{L}\,C$ or $S = \dfrac{T}{K}\,C$ where<br>$C = \dfrac{D}{1+\dfrac{\pi^2 D^4}{16A^2}}\left[1+0.15\left(\dfrac{\pi^2 D^4}{16A^2}-\dfrac{D}{2r}\right)\right]$,<br>where $D$ = diameter of largest inscribed circle<br>$r$ = radius of curvature of boundary at the point (*positive* for this case)<br>$A$ = area of the section |
| **17** I = section, flange thickness uniform.<br>$r$ = fillet radius, $D$ = diameter largest inscribed circle, $t = b$ if $b < d$, $t = d$ if $d < b$, $t_1 = b$ if $b > d$, $t_1 = d$ if $d > b$, | $K = 2\,K_1 + K_2 + 2\,\alpha\,D^4$ where<br>$K_1 = ab^3\left[\dfrac{1}{3}-0.21\,\dfrac{b}{a}\left(1-\dfrac{b^4}{12a^4}\right)\right]$<br>$K_2 = \dfrac{1}{3}\,cd^3$<br>$\alpha = \dfrac{t}{t_1}\left(0.15+0.1\,\dfrac{r}{b}\right)$ | |
| **18**  T-section, flange thickness uniform $r$, $D$, $t$ and $t_1$ as for Case 17. | $K = K_1 + K_2 + \alpha\,D^4$ where<br>$K_1 = ab^3\left[\dfrac{1}{3}-0.21\,\dfrac{b}{a}\left(1-\dfrac{b^4}{12\,a^4}\right)\right]$<br>$K_2 = cd^3\left[\dfrac{1}{3}-0.105\,\dfrac{d}{c}\left(1-\dfrac{d^4}{192c^4}\right)\right]$<br>$\alpha = \dfrac{t}{t_1}\left(0.15+0.10\,\dfrac{r}{b}\right)$ | At a point where the curvature is negative (boundary of section concave, or reentrant) this max. stress is given approximately by: |
| **19**  L-section,<br>$r$ and $D$ as for Cases 17 and 18.<br>$b \gtrless d$ | $K = K_1 + K_2 + \alpha\,D^4$ where<br>$K_1 = ab^3\left[\dfrac{1}{3}-0.21\,\dfrac{b}{a}\left(1-\dfrac{b^4}{12a^4}\right)\right]$<br>$K_2 = cd^3\left[\dfrac{1}{3}-0.105\,\dfrac{d}{c}\left(1-\dfrac{d^4}{192c^4}\right)\right]$<br>$\alpha = \dfrac{b}{d}\left(0.07+0.076\,\dfrac{r}{b}\right)$ | $S = G\dfrac{\theta}{L}\,C$ or $S = \dfrac{T}{K}\,C$<br>where $C = \dfrac{D}{1+\dfrac{\pi^2 D^4}{16A^2}}\left[1+\left\{0.118\,\log_e\left(1-\dfrac{D}{2r}\right)-0.238\,\dfrac{D}{2r}\right\}\tan h\,\dfrac{2\phi}{\pi}\right]$<br>where $D$, $A$ and $r$ have same meaning as before and $\phi$ = angle through which a tangent to the boundary rotates in turning or traveling around the reentrant portion, measured in radians. (Here $r$ is *negative*.) |
| **20**  U-section or Z-section. | $K$ = sum of $K$'s of constituent L-sections, computed as for Case 19. | The above formulas should also be used for Cases 12 and 13 when $t$ is relatively large compared with radius of median line. |

# MODERN MECHANICAL DESIGNS

## *Baldwin Locomotive for the New Haven*

In addition to being streamlined, the ten new steam locomotives for the New York, New Haven & Hartford Railroad have disk wheels, concealed smokestacks with smoke deflectors and numerous other new design features

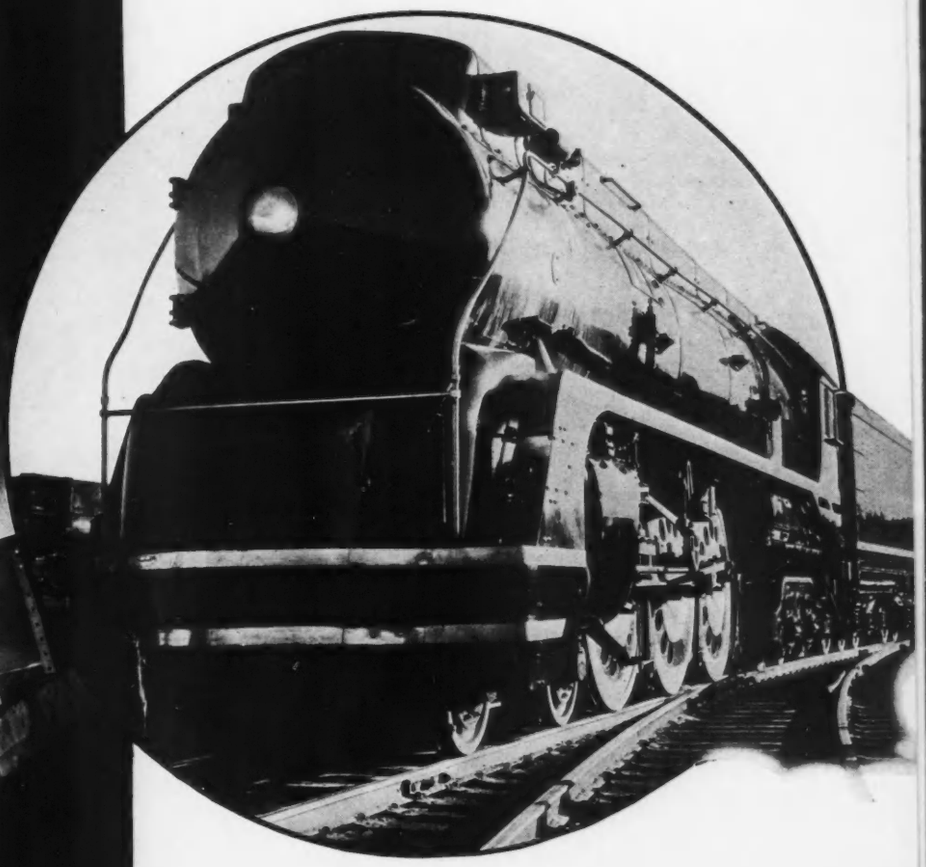


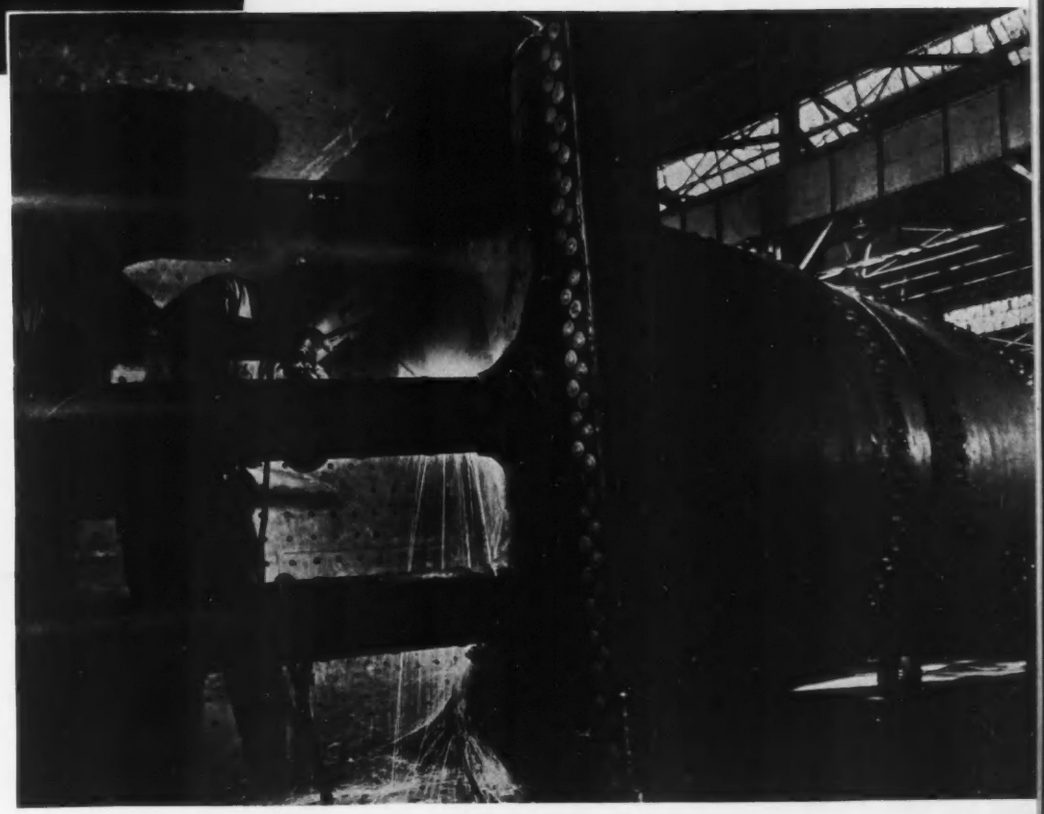
LOCOMOTIVE with tender is more than 97 ft. long, 10 ft. wide and 15 ft. 4 in. high. Completely equipped to run, the total weight is 694,500 lb., including the tender load of 16 tons of coal and 18,000 gal. of water, enough for a non-stop trip from New York to Boston.

Working steam pressure is 285 lb. per sq.in., giving initial tractive effort of 44,000 lb. Driving wheels are 80 in. diameter of a new disk type design. Maximum speed hauling a 15 car train is 80 mi. per hour.

Frames with cylinders, cross ties, bumper beams, cradle frames, air pump brackets and turbo injector brackets are all a one-piece 50,000 lb. steel casting, thereby eliminating more than 250 bolts or rivets.

The design called for a considerable amount of welding including the welding of the splash plates in the tender





tank and welding inside the firebox, illustrated above and to the right.

All locomotive axles are mounted on roller bearings. A lubricator driven off the valve linkage furnishes oil to all bearings and guides except the crankpins which are grease lubricated.

Sheathings over the fireboxes and boilers conceal the sandboxes, steam domes, safety valves, steam turrets and smokestacks, leaving long unbroken lines from the cabs to the front of the boilers. The cabs themselves have been streamlined into the fireboxes and boilers, with only the engineer's window breaking the lines. The front of the boiler comes out into a cone, with the headlight forming the apex of the cone. Along each side of the locomotive one stainless steel strip running the entire length, and two shorter strips over the leading trucks, further accentuate the streamlining.

# *Standards for Spring Design–II*

W. M. GRIFFITH
*Product Engineer, Atlas Imperial Diesel Engine Company*

## Allowable working stresses based on endurance limits of the materials; also design formulas incorporating the Wahl correction factor

IN LINE with the latest developments in the design of springs, it was obvious that endurance limit charts had to be drawn up, from which the allowable working stresses for Class 1 springs could be selected. A great deal of time was spent in investigating and reviewing all available literature on the subject, the final results being the endurance limit chart as given in Figs. 6 to 10. The endurance limits indicated are based on the ultimate tensile properties as given by the charts, Figs. 1 to 5 in Part I of this article.

As in the case of the tensile and torsion properties of the material, it was deemed advisable to permit less than the maximum stress indicated on the endurance chart in their application to the design of springs for use in Class 1 service. This was necessary because of the reduction of 5 per cent already made in the ultimate tensile properties as covered in the material specification sheets and, in addition, to provide a sufficient factor of safety to insure the designing and securing of springs which would not fail in service. To determine how closely the elastic limits could be approached, the investigation covered the amount of set in compression springs as the elastic limit is approached and these values were taken into consideration in the preparation of the final endurance limits charts Figs. 6 to 10. On all of these charts the endurance limits are 80 per cent of the actual maximum values.

By reducing the permissible maximum working stress as described above, it is felt that a sufficient margin of safety is allowed to preclude the failure of springs from a design standpoint. These stresses, and those permitted for Class 2 and Class 3 service as given by Table II, are somewhat different from those covered by the original standard which appeared in the July, 1933 *Product Engineering* in that the Wahl stress factor, Fig. 11, is included in all of the formulas for stress calculation.

In order to properly cover the maximum stresses permissible in extension springs and torsion springs, the average ratio of the torsional properties to the tensile properties, as covered by the ma-

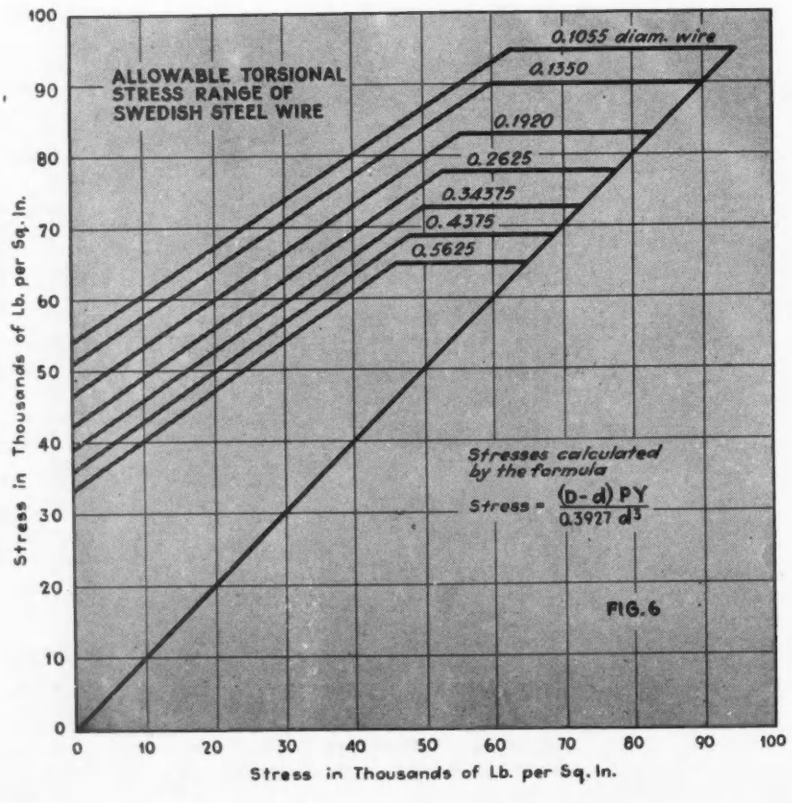



**TABLE II—MAXIMUM**

| Material<br>Gage<br>Modulus "G"<br>Modulus "E" | | Swedish Steel<br>Washburn & Moen<br>11,500,000<br>30,000,000 | | |
|---|---|---|---|---|
| Class | Size | Comp. | Exten. | Torsion |
| 1 | 0.0085 to 0.105 | | | |
| | 0.1055 to 0.2625 | | See | text |
| | 0.2812 to 0.5625 | | | |
| 2 | 0.0085 to 0.105 | | | |
| | 0.1055 to 0.2625 | 66,000 | 52,800 | 79,200 |
| | 0.2812 to 0.5625 | 55,250 | 44,200 | 66,300 |
| 3 | 0.0085 to 0.105 | | | |
| | 0.1055 to 0.2625 | 77,500 | 62,000 | 93,000 |
| | 0.2812 to 0.5625 | 65,000 | 52,000 | 78,000 |

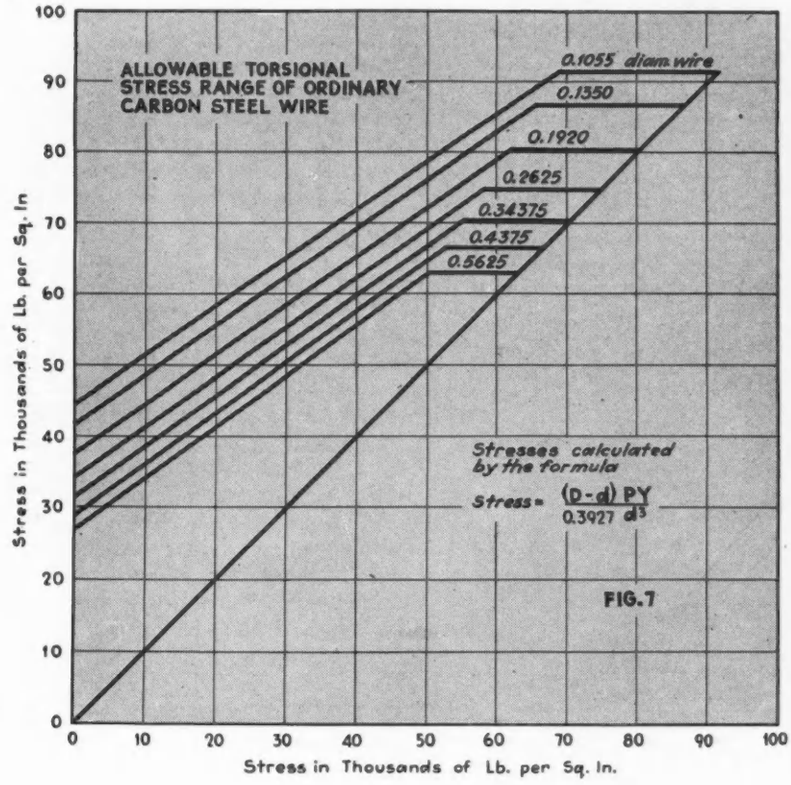


FIG. 7

terial specification sheets, was taken into account. Thus, these stresses could be based on the endurance limit charts with perfect safety. With reference to all Class 1 extension springs, the maximum permissible working stresses are limited to 75 per cent of the maximum values given by the endurance charts, and torsion springs are limited to 110 per cent of the values given by the endurance charts.

In Table III are given the equations and formulas for calculating compression, extension and torsion springs. Fig. 11 gives the curve for the Wahl factor for various values of spring index.

With reference to the spring index, it will be noted that when the spring index is less than 5, the Wahl factor increases rapidly. The optimum value for the spring index is 9, and this value should be approached by the designer as closely as possible.

For the equations given, the values to be used for the torsional modulus of elasticity and the maximum permissible stresses for the various services for Class 2 and Class 3 springs are given in Table II. The maximum permissible working stresses for all Class 1 springs must not exceed the limits as given by the endurance charts, as explained above. In addition, the springs shall be designed so that their natural frequency is of sufficient magnitude to escape resonance with any harmonic below the twentieth order. That is:

$$\frac{250\ d \times \sqrt{G}}{(D-d)^2 N} = 20$$

Deflection cycles per min.

The order of harmonic as calculated by this equation should be

## PERMISSIBLE STRESSES

| Carbon Steel Washburn & Moen 11,500,000 30,000,000 | | | Vanadium Steel Washburn & Moen 11,500,000 30,000,000 | | |
|---|---|---|---|---|---|
| Comp. | Exten. | Torsion | Comp. | Exten. | Torsion |
| | | | | | |
| | See | text | | See | text |
| | | | | | |
| | | | | | |
| 63,750 | 51,000 | 76,500 | 76,000 | 60,800 | 91,200 |
| 53,500 | 42,800 | 64,200 | 64,250 | 51,400 | 77,100 |
| | | | | | |
| 75,000 | 60,000 | 90,000 | 89,500 | 71,600 | 107,400 |
| 63,000 | 50,400 | 75,600 | 75,500 | 60,400 | 90,600 |

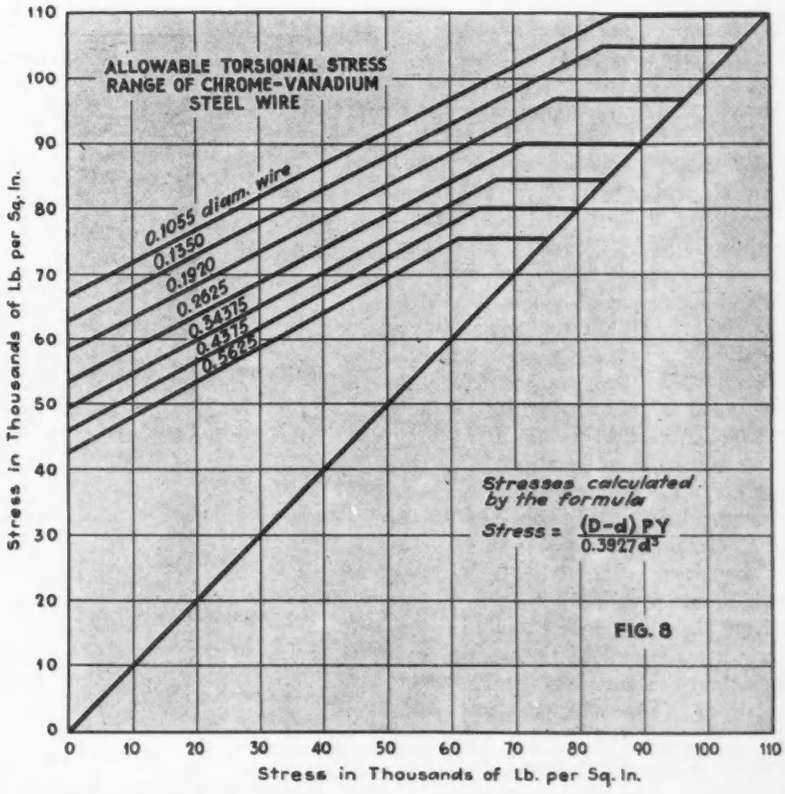


FIG. 8

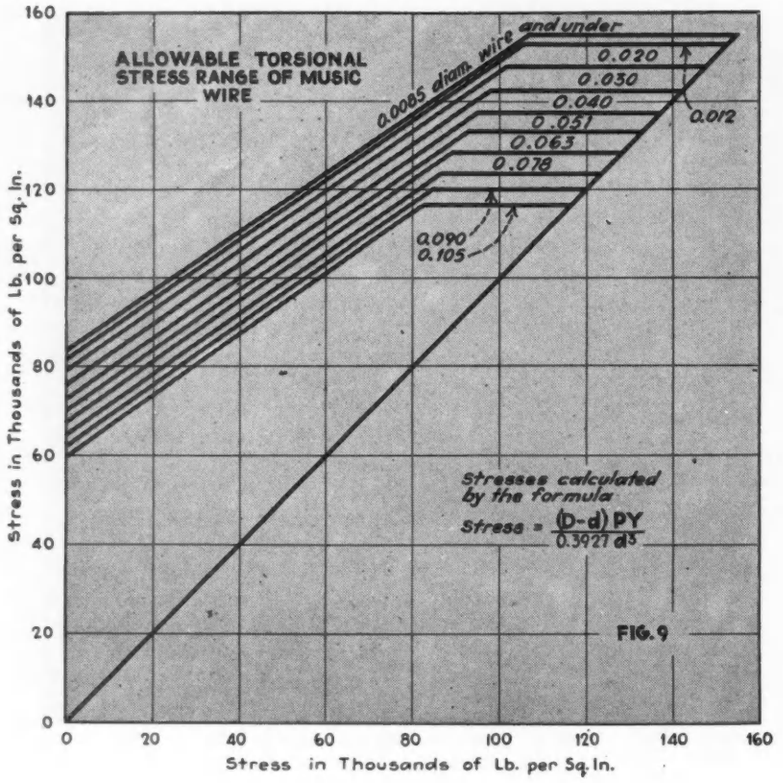



FIG. 9

as much above 20 as possible. In this equation $d$ is the diameter of the wire, $D$ is the outside diameter of the coil, $N$ is the number of active turns, $G$ is the modulus of elasticity in torsion, and the right-hand member of the equation gives the order of the harmonic.

The use of the endurance chart is best explained by an example. A valve spring made of Swedish steel of wire 0.177 in. diam. is stressed from 62,000 lb. per sq. in., with the valve closed and then up to 81,000 lb. per sq. in. when the valve is open, as calculated by the formula. It is required to determine if this stress range is permissible. Referring to the endurance chart, Fig. 6, starting from 62.0 on the lower stress line and proceeding in a vertical direction, to the line representing the next wire diameter larger than 0.177 in., which is 0.1920 in., the maximum stress is read from the scale on the left of the chart, the reading being 83.0, or 83,000 lb. per sq. in. The given stress range is therefore safe.

The compression spring formulas given here are based on $1\frac{1}{4}$ dead coils at each end of the spring, and less than this number should not be assumed. The springs must have the ends ground square with the center line. The minimum and maximum inside or outside diameter will be determined by the designer with the restriction that the spring must clear all adjacent parts when compressed. Springs should be guided at both ends on the outside or inside or both. All compression springs should be wound right hand except where they operate inside one another, in which case they should be wound one right hand and the other left hand. Minimum working length of the spring determines the position of the spring beyond which additional compression is not permitted. This length, therefore, is a definite part of the dimensions which control the spring characteristics and does not necessarily indicate the position of the spring in its application.

Extension springs, designed in accordance with the formulas given herewith, may be close wound, with or without initial tension, or they may be open wound, with the exception that they should always be wound without initial tension when the load capacity is an important factor. The dimensions of extension springs shall be fixed so as to clear all adjacent parts when the spring is extended. All extension springs shall be wound right hand except where required otherwise. It should be noted that the maximum working length determines the position of the spring beyond which additional extension is not permitted. This length, therefore, is a definite part of the dimensions which control the spring characteristics and does not necessarily indicate the position of the spring in its application.

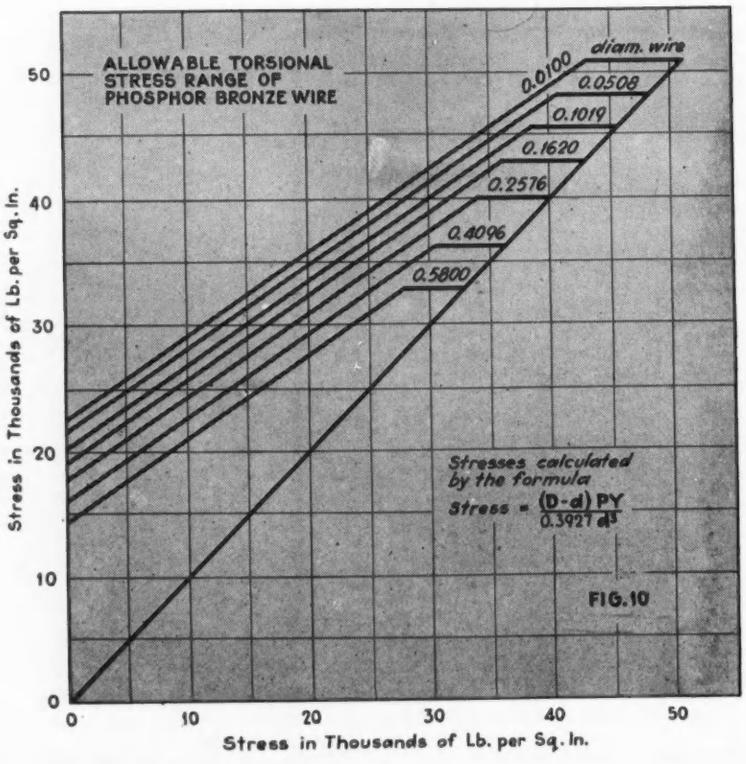



FIG. 10

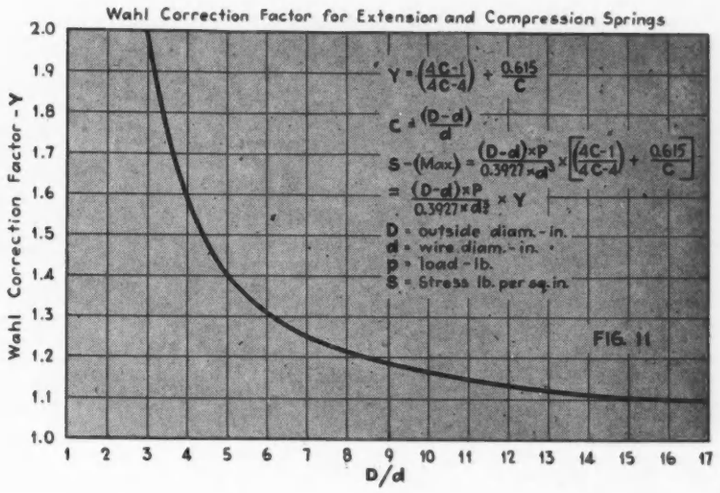
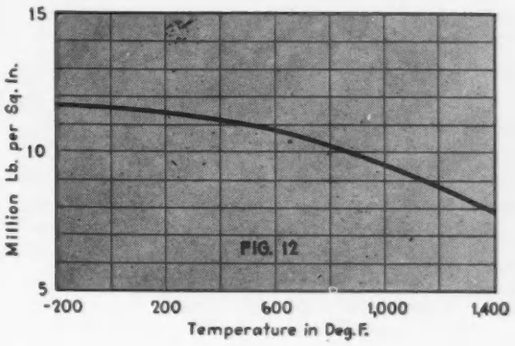

Wahl Correction Factor for Extension and Compression Springs
Wahl Correction Factor - Y
D/d
Y = (4C-1)/(4C-4) + 0.615/C
C = (D-d)/d
S = (Max.) = (D-d)×P/(0.3927×d³) × [(4C-1)/(4C-4) + 0.615/C]
= (D-d)×P/(0.3927×d³) × Y
D = outside diam. - in.
d = wire diam. - in.
P = load - lb.
S = Stress lb. per sq. in.
FIG. 11

***Fig. 11—Chart giving the Wahl factor used in calculating the stress in extension and compression springs.***

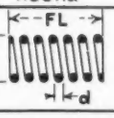
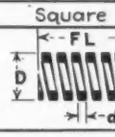
***Fig. 12—Variation of the elastic modulus of elasticity of steel with temperature***




# Table III—Spring Formulas

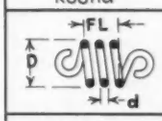
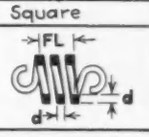
## COMPRESSION SPRING FORMULAS

Spring index = $\frac{D}{d}$ or $\frac{D}{b}$ = 5 - (minimum)

| Round | Square | Rectangular |
|---|---|---|
|  |  | |
| $W = \frac{0.3927 \times S \times d^3}{(D-d) \times Y}$ | $W = \frac{0.444 \times S \times d^3}{(D-d) \times Y}$ | $W = \frac{S \times b \times t \times \sqrt{b^2+t^2}}{3.185 \times (D-b) \times Y}$ |
| $N = \frac{MWL-(2.25 \times d)}{d \times 1.10}$ (max.) | $N = \frac{MWL-\left[d \times \left(\frac{D}{D-d}+1\right)\right]}{0.53 \times d \times \left(\frac{D}{D-d}+1\right)}$ (max.) | $N = \frac{MWL-\left[t \times \left(\frac{D}{D-b}+1\right)\right]}{0.53 \times t \times \left(\frac{D}{D-b}+1\right)}$ (max.) |
| $F = \frac{8 \times P \times (D-d)^3}{G \times d^4}$ | $F = \frac{5.58 \times P \times (D-d)^3}{G \times d^4}$ | $F = \frac{11.16 \times P \times (D-b)^3}{G \times b \times t \times (b^2+t^2)}$ |
| $F_N = F \times N$ | $F_N = F \times N$ | $F_N = F \times N$ |
| $FL = F_N + MWL$ | $FL = F_N + MWL$ | $F_L = F_N + MWL$ |
| $Pitch = \frac{FL-(2.25 \times d)}{N}$ | $Pitch = \frac{FL-\left[d \times \left(\frac{D}{D-d}+1\right)\right]}{N}$ | $Pitch = \frac{FL-\left[t \times \left(\frac{D}{D-b}+1\right)\right]}{N}$ |
| Load per inch of deflection = $\frac{P}{F_N}$ | Load per inch of deflection = $\frac{P}{F_N}$ | Load per inch of deflection = $\frac{P}{F_N}$ |
| Solid length = $(N+2.25) \times d$, | Solid length = $\left[0.48 \times d \times \left(\frac{D}{D-d}+1\right) \times N\right] + \left[d \times \left(\frac{D}{D-d}+1\right)\right]$ | Solid length = $\left[0.48 \times t \times \left(\frac{D}{D-b}+1\right) \times N\right] + \left[t \times \left(\frac{D}{D-b}+1\right)\right]$ |
| $MWL = (d \times 1.10 \times N) + (2.25 \times d)$ | $MWL = \left[0.53 \times d \times \left(\frac{D}{D-d}+1\right) \times N\right] + \left[d \times \left(\frac{D}{D-d}+1\right)\right]$ | $MWL = \left[0.53 \times t \times \left(\frac{D}{D-b}+1\right) \times N\right] + \left[t \times \left(\frac{D}{D-b}+1\right)\right]$ |

## EXTENSION SPRING FORMULAS

$F_N = F \times N$ Pitch = $\frac{FL}{N}$ Load per in. deflection = $\frac{P}{F_N}$

| Round | Square | Rectangular |
|---|---|---|
|  |  | |
| $W = \frac{0.3927 \times S \times d^3}{(D-d) \times Y}$ | $W = \frac{0.444 \times S \times d^3}{(D-d) \times Y}$ | $W = \frac{S \times b \times t \times \sqrt{b^2+t^2}}{Y \times 3.185 \times (D-b)}$ |
| $N = \frac{FL}{d}$ (max.) | $N = \frac{FL}{0.48 \times d \times \left(\frac{D}{D-d}+1\right)}$ | $N = \frac{FL}{0.48 \times t \times \left(\frac{D}{D-b}+1\right)}$ |
| $F = \frac{8 \times P \times (D-d)^3}{G \times d^4}$ | $F = \frac{5.58 \times P \times (D-d)^3}{G \times d^4}$ | $F = \frac{11.16 \times P \times (D-b)^3}{G \times b \times t \times (b^2+t^2)}$ |

## TORSION SPRING FORMULAS

Pitch = $\frac{FL}{N}$

| Round | Square or Rectangular |
|---|---|
| $W = \frac{0.098 \times S \times d^3}{R}$ | $W = \frac{0.166 \times S \times t \times b^2}{R}$ |
| $F_N = \frac{20 \times P \times l \times R^2}{E \times d^4}$ | $F_N = \frac{12 \times P \times l \times R^2}{E \times b^3 \times t}$ |
| $N = \frac{FL}{d}$ (max.) | $N = \frac{FL}{0.48 \times t \times \left(\frac{D}{D-b}+1\right)}$ (max.) |

E = *Modulus of elasticity*
F = *Deflection of 1-turn, inches*
$F_N$ = *Total deflection, inches*
FL = *Free length, inches*
G = *Torsional modulus of elasticity*
l = *Length of rod (effective spring length uncoiled), inches*
MWL = *Minimum working length, inches*
N = *Number of effective turns*
P = *Load, pounds*
R = *Length of arm, inches*
S = *Stress, lb. per sq. in.*
W = *Carrying capacity, pounds.*
Y = *Wahl factor*

# *Convoluted Flexible Metal Hose*

ELBERT E. HUSTED and DAVID INGALLS
*Titeflex Metal Hose Company*

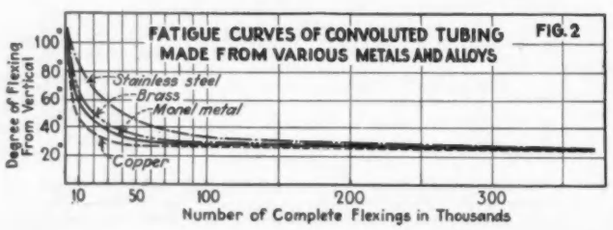
**Ways to install and to calculate tubing lengths for proper flexing and radial movement, comparative fatigue life of metals used, and data on head lost in fluid flow**

THE TERM "flexible metal tubing" is often misinterpreted. It is true that all metals are flexible to a limited extent, but in no sense do the metals out of which tubing is fabricated possess any special properties of flexibility.

Flexibility of spirally wound tubing is obtained in two ways. One method allows movement in the joints. The other method makes use of a longitudinal displacement of convolutions formed in the walls of the tube. Each type has its special advantages and is made up in many styles and sizes to meet the diversified demands of the automotive, aviation, railroad and industrial fields. The controlling factors in design are: lightness, durability, and cost.

Convoluted tubing, a cross-section of which is shown in Fig. 1 derives its flexibility from its wall construction. The convolutions are formed in a way which, when the tubing is bent, allows enough longitudinal displacement to take up the difference in length of curvature between the outside and the inside edges without deforming or stretching the metal. The only action that takes place in the metal is a slight bending in the walls of the convolution. The resulting stresses thus set up remain within the elastic range of the metal and, therefore, do not bring about any permanent deformation of the tube.

If one end of a ¼ in. copper tube, say 4 ft. long, were fastened in a fixed position, the other end could then be oscillated backwards and forwards for a distance of ½ in. indefinitely, without causing a work hardening or fatigue of the tube. If the same thing were done to a one foot length of copper tube it would not be long before it would fatigue and break. Now by convoluting the 4 ft. length of copper tube, it would shorten to approximately a one foot length of flexible tubing, which could be flexed backward and forward indefinitely the same as the original four foot length. The stresses in the one foot length of flexible tubing would



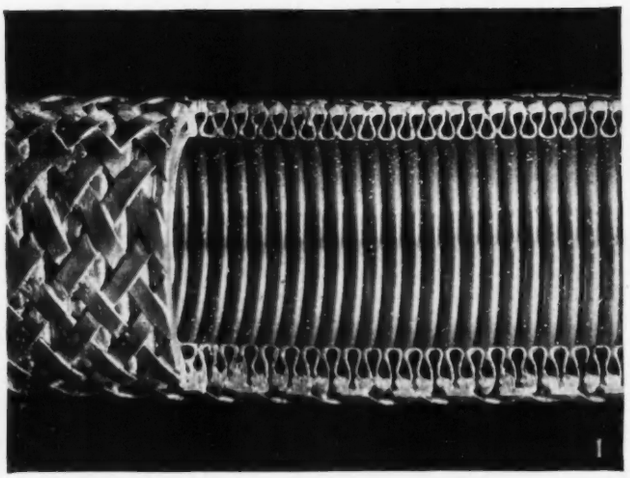



FIG. 3

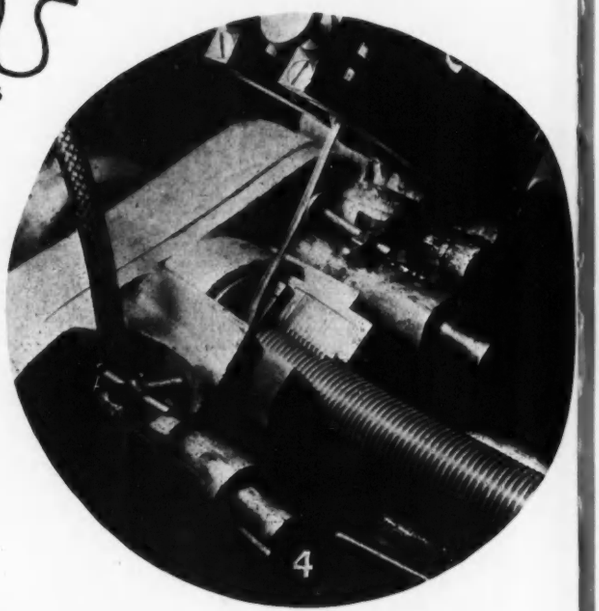



***Fig. 1—Cut-away section of spirally wound flexible metal tubing, formed to maintain uniform wall thickness without straining the metal. Fig. 2—Comparative fatigue curves of convoluted flexible tubing made from various metals and alloys. Fig. 3—Section at the all-metal joint with four thicknesses of metal. Fig. 4—Forming the all-metal inner core of flexible pressure tubing***

be distributed over the same amount of metal as in the 4 foot length of solid tube. It is this principle of keeping the stresses below the fatigue limit of the metal that makes flexible tubing possible.

The degree to which the convoluted tubing can flex is dependent upon the following factors: Height of the convolutions with respect to the I.D. of the tube; number of convolutions per inch of length; space between each convolution; curvature of the convolutions; thickness of metal; uniformity of cross-section of the metal; grain size and metal composition. All of these factors are taken into consideration in the design to arrive at a balanced proportion for each size. The gage of the metal is kept as thin as possible, so as to reduce the stress differentials to a minimum in the bending action. The limit is determined by the minimum stiffness required in the walls of the convolutions.

Convoluted tubing is available with the inner core fabricated out of brass, stainless steel or Monel metal. The physical properties of the various metals out of which the tubing is fabricated are by no means the same. It has been found that they also exhibit marked differences in properties after they have been made

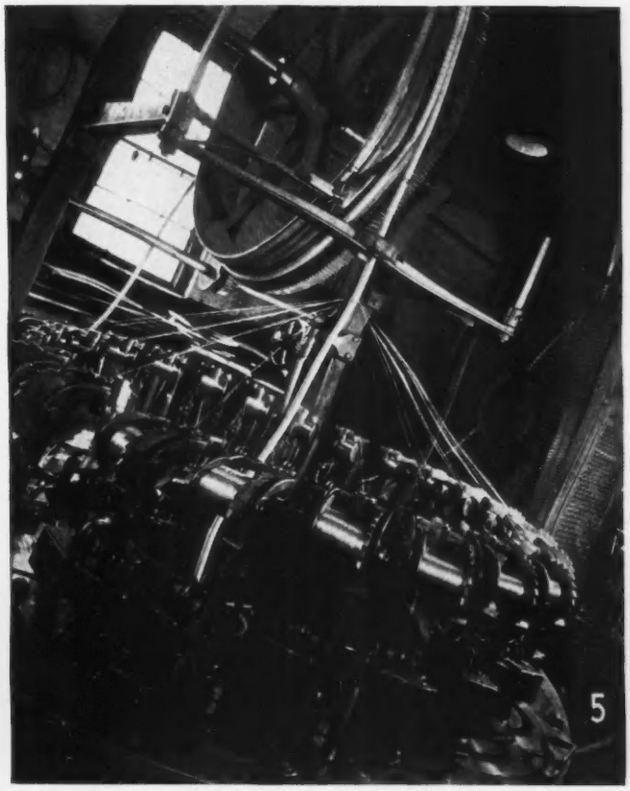



***Fig. 5—Small braiding machine covering a flexible metal tube with bronze braid***

into tubing, both in respect to fatigue life and resistance to corrosion. The graph in Fig. 2 shows the comparative merits of each metal with regard to fatigue.

Convoluted tubing is fabricated by winding spirally a profiled metal strip. The edges of the strip are shaped to form a double locked fixed seam at the outer circumference of the tube wall. The section of the all-metal joint is shown in Fig. 3. In Fig. 4 is shown the operation of winding and spinning the convoluted inner core of the tubing.

Tubing of this type is always protected and supported by a metal braid which adds strength to the inner core, prevents elongation when the tubing is subjected to pressure, protects the tubing against abrasion in handling and service, restricts the bending to definite limits and resists twisting forces that set up torsional stresses in the inner core. The braided coverings consist of tubular braids of a flat narrow bronze strip or a basket woven casing of fine bronze wire. The amount and the arrangement of the braiding construction depends entirely on the requirements of the intended service.

When confronted with an unusual flexible tubing problem, it is well to consult the manufacturer as to type of construction best suited for the service required. In this way the user is assured of the best possible assembly for any particular problem and receives the benefit of the engineering knowledge and experience of the manufacturer who specializes in this field.

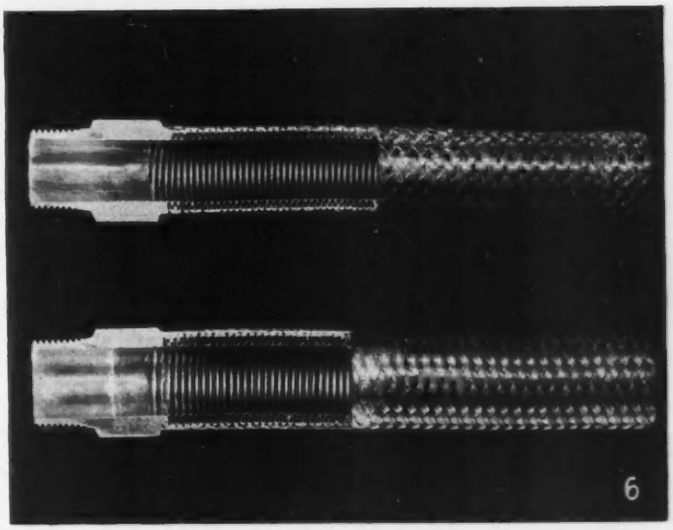
The small braiding machine shown in Fig. 5 is covering the inner core of a tube with a wire braid.

Because of the diversified fields in which flexible metal tubing is used, practically every type of copper tube, pipe thread, and streamline sweated fitting in common use today is available. Fittings are made of brass unless otherwise specified. Usually they are connected as shown in Fig. 6, by inserting the tube into a counterbore in the back of the fitting and then filling with soft solder. A threaded insert type of fitting is used, as shown in Fig. 7, where high steam pressures are involved. This is screwed on to the tubing and is then filled in with a high lead content solder. This fitting cannot be blown off under pressure, even at temperatures in excess of the melting point of the solder used.

For extremely high temperature work, a fitting such as is shown in Fig. 8 is used, the tubing is silver soldered to the back sleeve. A streamlined sweated fitting with a section as shown in Fig. 9 can be supplied. These fittings can withstand the heat needed for soldering and sweating without danger of damaging the connection at the tubing.

In applications of tubing where the movement is reciprocating, the line should be installed so that it

***Fig. 6—Deep uniform convolutions permit great flexibility. Single flat ribbon and double fine wire braid are shown. Note the secure bond between the coupling and the tubing***



forms a "U" with the fixed end at mid-position of the stroke. The moving end should be connected so that it points in the direction of the motion. The location of the two ends should be based on the radius of bend, as given in Table I, for the best working conditions. Sometimes it is impossible to comply with these dimensions; however, they should be approached and adhered to as closely as possible.

After the positions of the two ends have been determined, the length of the tubing required can be calculated as follows: The length of stroke plus sixteen times the I.D. of hose, plus one and one-half times the distance apart of the two ends will equal the overall length needed.

For a radial movement about some pivot point, such as on a laundry pressing machine as shown in Fig. 10,

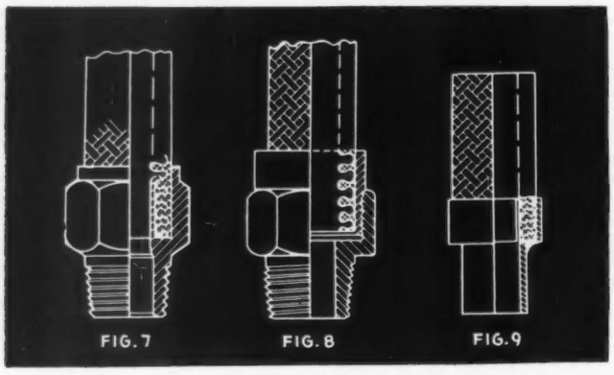



***Fig. 7—Type of end fitting for commercial hose. Fig. 8—Type of end fitting used for high-pressure steam and high temperature service. Fig. 9—Type of streamlined end fitting***

the tubing should be installed so that its angular displacement is just one-half of the angular displacement of that part of the machine to which it is connected. If this is done, there will be no tendency for the line to twist. The length of tubing required depends upon distance between centers and always should be long enough to permit a well rounded bend with a slight straightening out of the line at each end.

In addition to these general rules, sharp bends should not be used, particularly at the coupling where stress concentrations frequently occur and bring about premature failures. It is important to bear in mind at all times that it is magnitude of movements, not frequency, that should be avoided as much as possible when using convoluted tubing.

A common belief exists that the resistance to flow in flexible tubing is quite high. Tests prove conclusively that such is not the case and that oversizing retards the flow more than does tubing of the correct size. When flowing through flexible metal tubing the liquid fills up the convolutions and forms a liquid wall to flow through, thus reducing the resistance to a minimum.

As a guide for predicting the probable resistance to flow through flexible tubing these formulas are presented, where

$H$ = head loss expressed in ft.
$V$ = velocity in ft. per sec.
$D$ = inside diam. of tubing expressed in ft.
$g$ = 32.2 ft. per sec.$^2$, acceleration of gravity
$F$ = velocity factor
$C$ = area ratio factor, this is ratio of the area of hole in coupling to the area of hole in iron pipe or copper tube
$B$ = bend angle factor

Friction loss in line, $H = FLV^2/D\ 2\ g$
where $F$ is 0.04 when $V$ equals 1 ft. per sec.,
$F$ is 0.06 when $V$ equals 5 ft. per sec.,
and $F$ is 0.08 when $V$ equals 10 ft. per sec.

Loss at coupling, $H = CV^2/2g$
where $C$ is 1.0 when area ratio equals 0.1,
$C$ is 0.5 when area ratio equals 0.5,
and $C$ is 0.1 when area ratio equals 1.0

Loss at bends, $H = BV^2/2g$
where $B$ is 0.5 for a 90 deg. bend,
and $B$ is 1.0 for a 180 deg. bend.

The many variables which come into play make accurate calculations impossible, and where conditions of flow are important, a flow test is recommended.

***Fig. 10 — Method of installing flexible metal hose for radial movement applications***

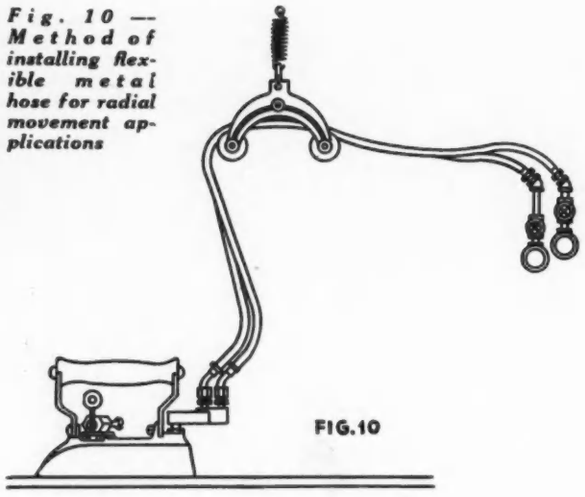



***TABLE I—CONVOLUTED FLEXIBLE TUBING DATA***

| Approx. Inside Diam. in In. | *Outside Diam. in In. | Weight Lb. per Foot | Radius of Bend in In. | Recommended Maximum Pressures Lb. per Sq.In. |
|---|---|---|---|---|
| 5/32 single braid | 0.334 | 0.10 | 2 | 2500 |
| 5/32 double " | 0.386 | 0.13 | 2 | 3000 |
| 3/16 single " | 0.367 | 0.14 | 3 | 2500 |
| 3/16 double " | 0.419 | 0.17 | 3 | 3000 |
| 1/4 single " | 0.433 | 0.15 | 4 | 2000 |
| 1/4 double " | 0.485 | 0.19 | 4 | 2500 |
| 5/16 single " | 0.508 | 0.21 | 4 | 1500 |
| 5/16 double " | 0.560 | 0.26 | 4 | 1800 |
| 3/8 single " | 0.588 | 0.26 | 5 | 1200 |
| 3/8 double " | 0.640 | 0.34 | 5 | 1400 |
| 1/2 single " | 0.789 | 0.52 | 6 | 1100 |
| 1/2 double " | 0.841 | 0.64 | 6 | 1200 |
| 5/8 double " | 0.966 | 0.74 | 7 | 800 |
| 3/4 single " | 1.075 | 0.67 | 9 | 700 |
| 3/4 double " | 1.127 | 0.85 | 9 | 800 |
| 1 single " | 1.387 | 0.88 | 11 | 500 |
| 1 double " | 1.439 | 1.09 | 11 | 600 |
| 1¼ single " | 1.699 | 1.06 | 12 | 350 |
| 1¼ double " | 1.751 | 1.45 | 12 | 400 |
| 1½ single " | 2.070 | 1.50 | 13 | 250 |
| 1½ double " | 2.122 | 1.82 | 13 | 300 |
| 2 single " | 2.570 | 1.85 | 15 | 200 |
| 2 double " | 2.622 | 2.24 | 15 | 250 |
| 2½ single " | 3.241 | 2.51 | 17 | 150 |
| 2½ double " | 3.301 | 2.96 | 17 | 200 |
| 3 double " | 3.817 | 3.50 | 18 | 100 |

* Outside diameters for sizes 5/32 to 1¼ in. inclusive are plus or minus 0.010 in. for sizes 1¼ to 3 inches are plus or minus 0.015 in.

# QUESTION AND COMMENT

## *Hydraulic Pressure Drop in Long Tubes*

● In answer to several questions raised by a reader, Mr. Hans Ernst presents the following information extending his discussion of the Poiseuille formula which was used in his article "Modern Hydraulic Control Systems and Circuits—I" published in the April, 1935 issue of *Product Engineering:*

*To the Editor:*

● My first article on hydraulics pointed out that whenever streamline flow is encountered the pressure drop produced in a given pipe is directly proportional to the velocity. This is usually expressed by the equation, established by Poiseuille:

$$P = \frac{32\,\mu L v}{d^2} \qquad (1)$$

$P$ = pressure drop in lb. per sq.ft.
$\mu$ = absolute viscosity of the fluid in f.p.s. units
$d$ = diameter of the pipe in ft.
$v$ = average velocity of fluid through pipe in ft. per sec.

But the average velocity of the fluid is equal to the quantity $Q$, divided by the area $\pi d^2/4$. Hence we may rewrite Equation (1):

$$P = \frac{128\,\mu L Q}{\pi d^4} \qquad (2)$$

If $P$ is to be expressed in lb. per sq. in., $L$ in in., $d$ in in., and $Q$ in cu. in. per min., then:

$$P = 0.0047\,\frac{\mu L Q}{d^4} \qquad (3)$$

If we now divide through by $Q$, we obtain the relation:

$$\frac{P}{Q} = 0.0047\,\frac{\mu L}{d^4} \qquad (4)$$

Since the right-hand side of the equation depends only on the physical dimensions of the pipe and on the viscosity of the fluid, it may be designated by the resistance factor $R$ and the equation used analogously to Ohm's Law of electrical circuits; thus:

$$\frac{P}{Q} = R$$

or $$P = Q R$$

In the article, $\mu$ was defined as "absolute viscosity of the fluid in poises." This is not, literally, as correct as the definition used above, calling for "absolute viscosity in f.p.s. units" (or slugs per ft. sec.), since all of the other factors are given in f.p.s. units.

It is unfortunate that the f.p.s. unit of viscosity in the gravitational system (slugs per ft. sec.) does not have an individual name. As a result the name "poise" is sometimes loosely used to cover the unit in either system, and it is in this way that the above mistake occurred. In the stress of assembling this material for publication, the error was overlooked, and, strangely enough, was not noticed in subsequent checking. It has also been noted that, in the article, Equation (2) had $\pi$ omitted from the denominator, although it appeared in the original manuscript. However, in Equations (3) and (4) it was included in the coefficient.

It has been suggested that Equation (1) should have $g$ in the denominator. In view of the foregoing remarks, this is not the case. The f.p.s. equivalent of the poise is likewise a gravitational unit which is expressed in slugs per ft. sec.; this in itself contains the value of $g$. The error lies not in the form of the equation, but in our statement that the value of $\mu$ should be given in poises.

In our own application of these equations, we customarily use values of absolute viscosity in slugs per ft. sec., converting, where necessary, from the c.g.s. poises, by multiplying by the factor 0.002088.

A better understanding of this matter of units may be had by subjecting Equation (1) to dimensional analysis. As used in the equation, $\mu$ is a gravitational unit which, if expressed dimensionally, must be given as $M/LT$ (See Eschbach: "Handbook of Engineering Fundamentals," p. 6-03). Then expressing each of the units in terms of mass, length, and time, we may write the Poiseuille equation as follows:

$$\frac{M}{LT^2} = \frac{M}{LT} \times \frac{L}{1} \times \frac{L}{T} \times \frac{1}{L^2}$$

$$= \frac{M}{LT^2}$$

Thus the equation has the dimensions of a pressure per unit area, since

$$\frac{\text{Pressure}}{\text{Area}} = \frac{\text{Force}}{\text{Length}^2} = \frac{\text{Mass} \times \text{Acceleration}}{\text{Length}^2}$$

$$= \frac{M}{L^2} \times \frac{L}{T^2} = \frac{M}{LT^2}$$

The Poiseuille formula applies only to tubes which are relatively long with respect to their diameters and to cases where the Reynolds Number is usually well below 1,000. When correctly used, it does give values which check with practice as may be seen from the following examples:

*Case I*

Oil: Gargoyle D.T.E. light
Temperature: 90 deg. F.
Absolute viscosity ($\mu$): 0.000798 slugs per ft.-sec.
Tube length: 36 in.
Tube inside diameter: 0.054 in.
Quantity: 60 cu. in. per min.

Then $$P = \frac{0.0047\,\mu LQ}{d^4}$$

$$= \frac{0.0047 \times 0.000798 \times 36 \times 60}{0.0000085}$$

= 952 lb. per sq. in.

With the Reynolds Number in this case equal to 351, the observed value of the pressure drop was 900 lb. per sq. in. However, by computation it will be found that the calculated value would be exactly equal to the observed value if the diameter of the tube were 0.0547 in. instead of 0.0540 in. (as assumed), or if the actual temperature were 91 deg. F., instead of 90 deg. F. Thus this result is well within the limits of experimental error.

*Case II*

All conditions were the same as in Case I, except that the temperature was raised to 150 deg. F. This gives a viscosity of 0.000209 slugs per ft.-sec., and a Reynolds Number of 1,338. Then:

$$P = \frac{0.0047 \times 0.000209 \times 36 \times 60}{0.0000085}$$

= 249 lb. per sq.in.

The observed value in this case was 375 lb. per sq. in. This shows that at this value of the Reynolds Number the resistance to flow was not purely viscous but was partially kinetic. Here, evidently, we are not

in the region of pure streamline flow.

*Case III*

The conditions of temperature and viscosity as in Case II, but a tube length of 18 ft. and a quantity of 8 cu. in. per min., were used. This gives a Reynolds Number of 178. Then:

$$P = \frac{0.0047 \times 0.000209 \times 216 \times 8}{0.0000085}$$

= 199 lb. per sq.in.

In this case, 200 lb. per sq.in. was the observed value. —HANS ERNST
*Cincinnati Milling Machine Co.*

*To the Editor:*

● Mr. Ernst has clarified completely the points I questioned. I agree that the units do check and that they are units of pressure. However, it is unquestionably difficult for one who is not specializing in hydraulic design to get a result in hybrid units when he was expecting, from the original statement of the formula, a result in the more ordinarily used f.p.s. units. The use of the viscosity unit "slugs per ft. sec." is new to me. I don't enjoy seeing another unit but I shall now be able to keep it straight.
—H. N. B.
*Lancaster, Ohio*

[*Editor's Note:* We have in preparation an article discussing further the subject of viscosity, the units used and the methods of converting from one system of units to another.]

# *Figuring Acceleration Time*

● In solving various product-design problems, it is often necessary for the engineer to calculate the time required for rotating machinery to accelerate or decelerate through a given range of speed. For example, he may wish to predetermine how long it will take some machine or group of machines, in the course of a proposed cycle of operations, to come up to the proper speed at which a certain operation may be started and then, upon completion of that operation, return to the initial speed, or to rest, ready for the next operation in the sequence.

In the following paragraphs a simple method of calculating the time required for acceleration or deceleration is presented and an example of such a calculation discussed.

If the accelerating or decelerating torque applied to the rotating machinery is constant throughout the speed range being considered, the time required can be readily calculated by means of the formula:

$$\text{Time in sec.} = \frac{WR^2\,(N_2 - N_1)}{308} \times \frac{1}{T}$$

Where $WR^2$ = weight in lb. × (radius of gyration in ft.)² of the rotating parts

$N_1$ = the lower speed in revolutions per min.

$N_2$ = the higher speed in revolutions per min.

$T$ = accelerating or decelerating torque in lb.-ft.

In cases involving an electric-motor-driven machine, it is convenient to express the torque in per cent of full-load torque (corresponding to a certain horsepower and speed rating), and the above formula then becomes:

$$\text{Time in sec.} = \frac{0.62WR^2\,(N_2 - N_1) \times \textit{Full-load Speed} \times 100}{(\textit{Full-load hp.} \times 1000 \times 1000 \times \textit{Per cent Full-load Torque})}$$

Although in many problems the accelerating or decelerating torque instead of remaining constant varies over the speed range involved, the formulas given above will still apply if the term $1/T$ is replaced by "$(1/T)$ average." The latter term represents the average, over the speed range considered, of the reciprocal of the torque. Thus it is necessary to calculate the reciprocal of the accelerating or decelerating torque at a sufficient number of speed points to plot a curve of this reciprocal and determine its average value. As will be shown in an example given later, there may be appreciable error in merely determining the average torque and substituting this in the formula instead of calculating the average of the reciprocal of the torque.

Frequently the accelerating or decelerating torque, over a considerable range of speed, varies as a straight line with the speed. When it does, the accelerating or decelerating time for the speed range can be calculated, using one of the formulas given above, by substituting "equivalent torque" for the torque $T$. To determine the equivalent torque, it is necessary merely to multiply the average torque for the speed range by the factor $K$, which may be obtained by referring to Fig. 1. There, the factor $K$ is shown plotted against the ratio $(T_1/T_2)$, where $T_1$ and $T_2$ are the highest and lowest values respectively of the accelerating or decelerating torque over the straight line section considered. The torque-speed curve can usually be represented with sufficient accuracy by a limited number of straight line sections. The accelerating or decelerating time for each section can be calculated by the method given, and these times added to give the total accelerating or decelerating time.

To illustrate this method, assume that a 100-hp., 1,750-r.p.m. wound-rotor induction motor is to accelerate a constant-torque load with a total $WR^2$ (for motor + load) of 500 lb.-ft.² Further, assume that the secondary control is designed to limit the torque peaks to 150 per cent of motor full-load torque and that the motor is allowed to accelerate on each step to a speed where the motor torque is 5 per cent greater than the load torque (See Fig. 2. The average accelerating torque is $(50 + 5)/2 = 27.5$ per cent. Since the ratio $T_1/T_2$ is 50/5 or 10 for each of the accelerating steps, the factor $K$, from Fig. 1, is 0.71 and the equivalent accelerating torque is 0.71 × 27.5 or 19.5 per cent. Since this

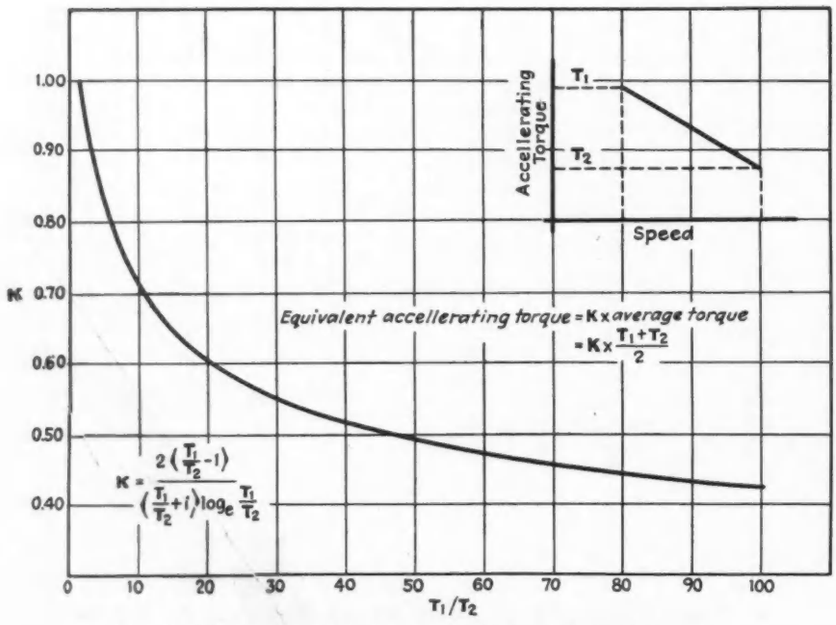



***Fig. 1—Curve for obtaining equivalent accelerating torque***

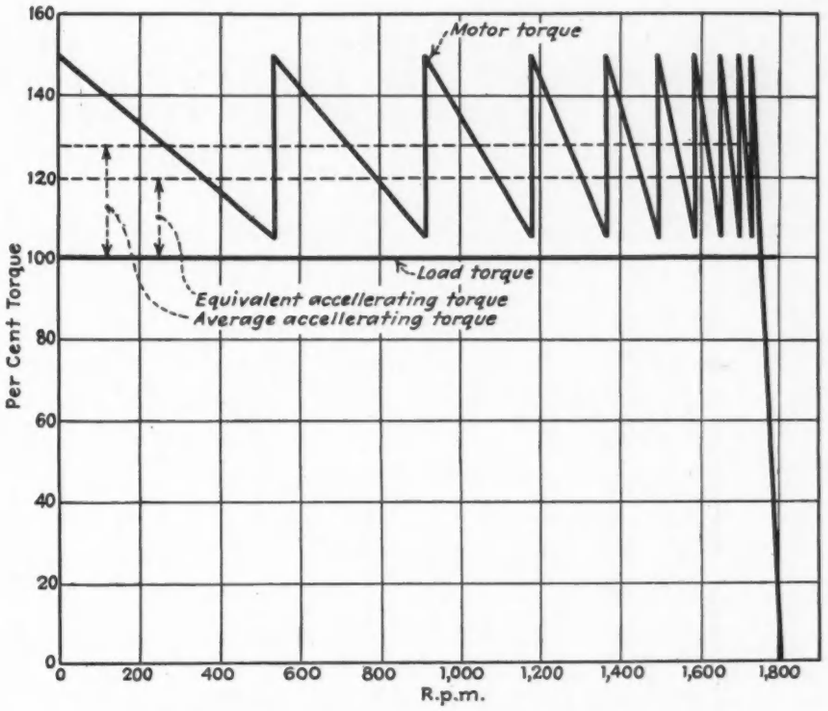



***Fig. 2—Speed-torque curve for a wound-rotor induction motor, showing effect of equivalent accelerating torque***

value is the same for each of the steps, the total accelerating time can be calculated as follows:

$$\text{Time in sec.} = 0.62 \times \frac{500}{100} \times \frac{1750 - 0}{1000} \times \frac{1750}{1000} \times \frac{100}{19.5} = 48.6$$

If the average accelerating torque (27.5 per cent) had been used directly in the formula, the calculated accelerating time would be 34.5 sec., an error of 29 per cent. If the equivalent accelerating torque had been different for each of the speed steps, it would have been necessary to calculate the accelerating time for each step and add to obtain the total accelerating time.

—M. N. HALBERG
*General Electric Company*

## *Discharge of Gear Pumps*

*To the Editor:*

● Recently I have seen several references to the theory of the gear type of pump in the Question and Comment pages of *Product Engineering*. It occurs to me that your readers might like to know of the article, "A Volumetric Displacement-Force Analysis of the Gear-Wheel Pump" prepared by W. H. Rasche, Professor of mechanism at the Virginia Polytechnic Institute.

—C. H. PARKER
*Baltimore, Md.*

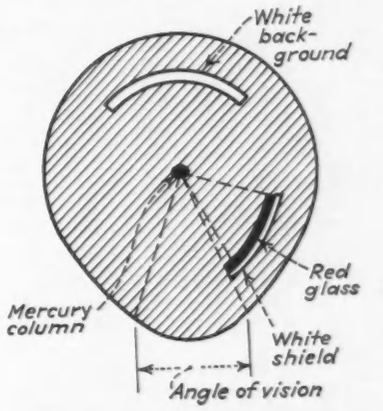
[*Editor's Note:* A pump having two 6-tooth, 30 deg. involute spur gears mounted in a conventional pump case was used as a basis for the calculations. Displacement analysis is discussed at some length, with problems and their detailed solutions. Mechanical efficiency of the impellers, force analysis and strength of teeth are also dealt with. Copies of this article which appeared in the "Virginia Tech Engineer," May, 1936, may be obtained at 25 cents each by addressing the Virginia Polytechnic Institute, Blacksburg, Va.]

## *Easily Read Thermometer Tube*

*To the Editor:*

● In the January number of *Product Engineering*, I was interested in seeing the short article on an easily read, three-lens thermometer tube. I recently saw the cross-section of another tube, as shown in the illustration, known as the Red-Reading-Mercury thermometer wherein the mercury column is red in color instead of being silvery. This is accomplished by means of reflection.

There is a strip of red glass backed by a white shield within the tube and because mercury acts as a mirror, the red glass is reflected onto the mercury column. When visioned through the lens at the front of the tube, only the red color is seen on the mercury column.




***Mercury column acting as mirror, reflects red from colored glass strip within thermometer***

—A. M. HOLLAND,
*New York, N. Y.*

# *Can You Work This One?*

H. E. SMITH
*Newark, N. J.*

***Solution to Last Month's Problem—***

### Bottle in the Stream

The problem can be solved more readily by reflection and calm analysis than by computation. Suppose a man were in a boat on a still mill pond and that he rowed for 30 minutes in one direction and then turned around and rowed back to the point from which he started. How long would it take him to cover the entire distance? Obviously. two times 30 minutes or one hour. A river may be considered as simply a mill pond in motion. During the hour that the man was rowing, the moving "mill pond" in the problem traveled two miles, as indicated by the distance the bottle moved. Hence, the stream was flowing at the rate of two miles an hour.

***This Month's Problem—***

### Chipmunk on the Log

A chipmunk jumps onto the end of a log ten feet long and three feet in circumference. His added weight starts the log rolling downhill. While the log rolls fifty feet down the hill, the chipmunk runs across to the other end of the log, always remaining on the top. How far does he travel?

# *Applications of Electrically*

Solenoid valves can be used in many different ways, this presentation being limited to a selected group of typical applications of the several different types of the valves. This is therefore not intended as a comprehensive list but rather as a digest of the more generally applicable methods of employing such devices, with the purpose of suggesting the numerous other ways in which it is possible to use electrically operated valves for both machine-control and industrial applications

1

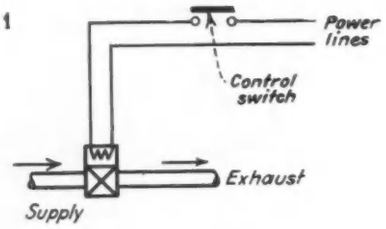



FIG. 1—Straight-way solenoid valve as commonly connected for simple fluid control. Control switch energizes solenoid, opening valve and permitting flow to begin.

2

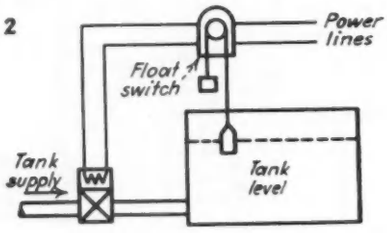



FIG. 2—Straight-way valve applied to automatically control liquid level. Float switch used as pilot control device for valve.

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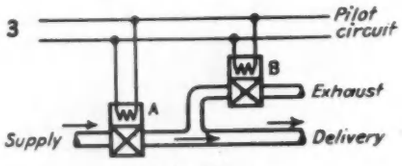


First condition
A-open, B-closed

FIG. 3—Two straight-way valves, *A* normally open and *B* normally closed, provide two-way fluid control. Energizing solenoids cuts off supply and vents delivery through exhaust.

4

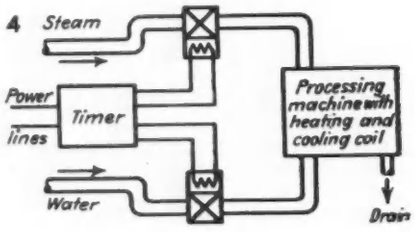


FIG. 4—Two straight-way valves offer means of automatically controlling cycle of processing machine, such as plastic molding press, having heating and cooling coils.

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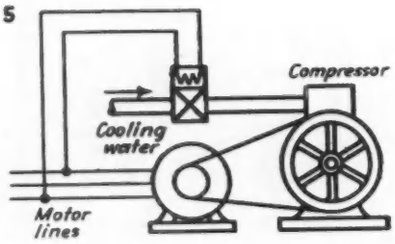
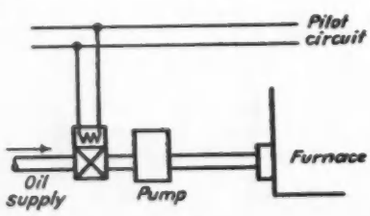



FIG. 5—Single straight-way valve can be connected across one phase of motor winding to start flow of cooling water to compressor whenever motor starts.

6

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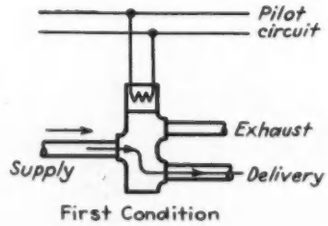
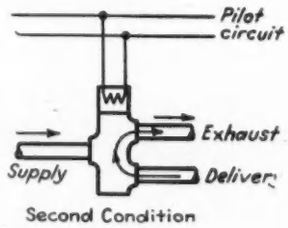



First Condition




Second Condition

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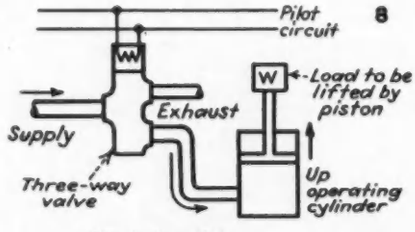
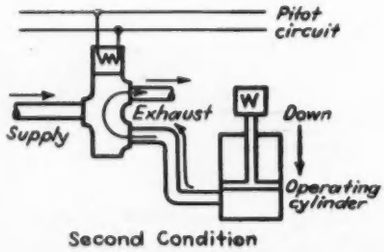



First Condition




Second Condition

# Operated Valves

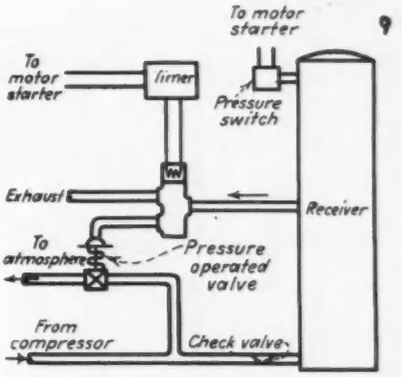



FIG. 6—Straight-way valve of trip type interlocked with oil-furnace control system to cut off oil supply upon loss of current to motor-driven pump or to atomizing equipment and upon occurrence of low water, low stack temperature, or similar conditions.

FIG. 7—Single three-way solenoid valve cuts off supply and vents delivery through exhaust. Application similar to that, shown in Fig. 3, using two straight-way valves.

FIG. 8—Three-way valve provides convenient means of controlling single-acting cylinders or diaphragms. Utilizing principle shown in Fig. 7, valve cuts off supply and vents delivery through exhaust, thus permitting return stroke of piston to take place.

FIG. 9—Three-way valve, used in manner similar to that shown in Fig. 8, unloads compressor until motor has time to come up to speed.

FIG. 10—Three-way valve arranged for multiple control of pressure-operated valves. In the first condition, Valve *A* is closed and Valve *B* open. In the second condition, the reverse is true, with Valve *A* open and Valve *B* closed.

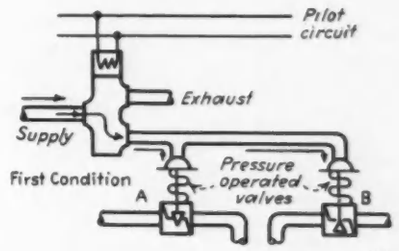
FIG. 11—Three-way valve applied as convenient means of transferring one supply to either of two deliveries.

FIG. 12—Three-way valve, utilizing inversion of principle shown in Fig. 11, offers means of transferring either of two supplies to a common delivery. Useful in applications where an emergency supply is provided.

FIG. 13—Four-way valve arranged to control double-acting cylinder. Upon energization of solenoid, operating rod of cylinder reverses direction.

FIG. 14—Four-way valve arrangement, employing principle shown in Fig. 13, provides automatic control of tank level through pressure-operated valve.

*Courtesy General Electric Co.*

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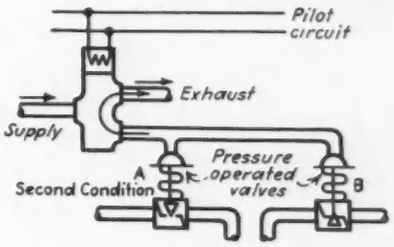


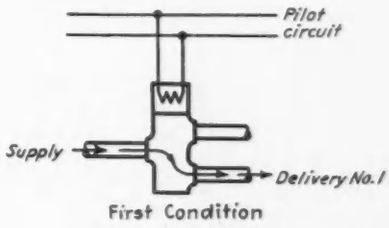


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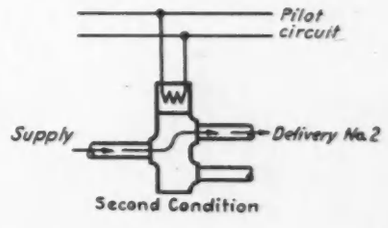


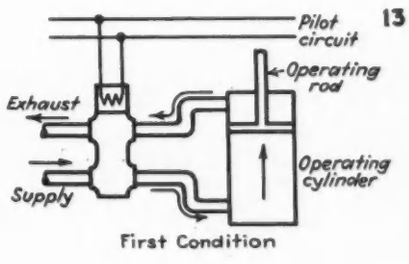


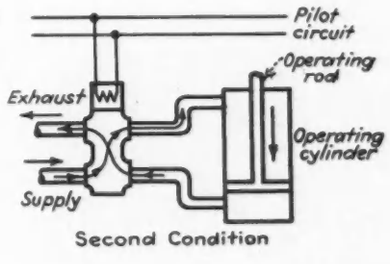


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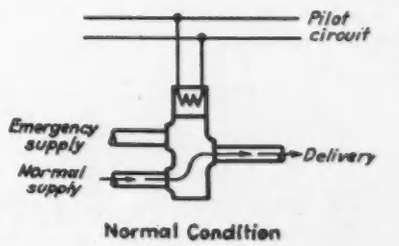
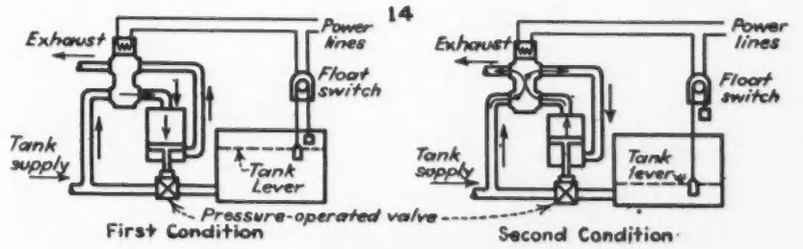


# NEWS

## *Roxalin Mobile Display to Tour Country*

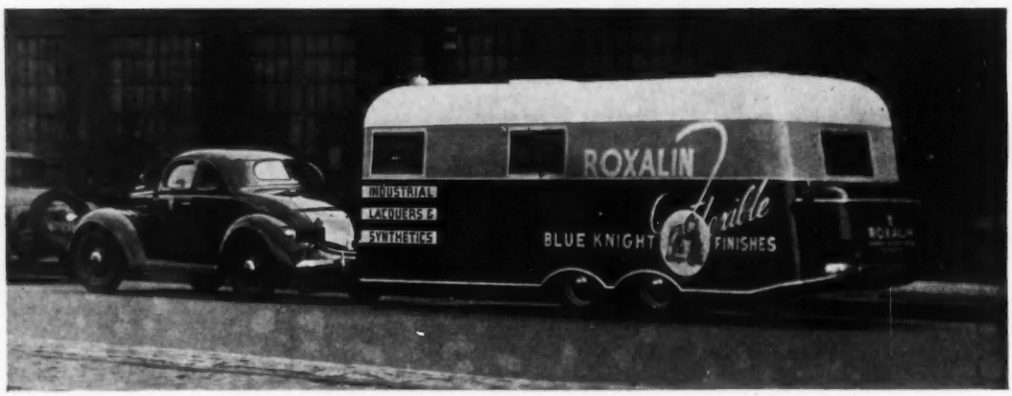
TO BE known as the Roxalin Trail Blazer, a 22 ft. elegantly equipped trailer with displays to demonstrate what lacquer and synthetic finishes can do, has been built by the Roxalin Flexible Lacquer Company. The displays consist of more than 200 parts and assembled products actually in commercial production which have been finished with Roxalin flexible lacquer and synthetic coatings in every conceivable style, design, color and shade, to stand every type of use and abuse, chemical and physical.

The base materials upon which finishes are shown include metals, wood, rubber, textiles, leather and paper. The finishing problems demonstrated are infinitely varied: resistance to human perspiration; resistance to weathering; resistance to chemicals—acids and alkalies; resistance to abrasion, blanking and forming. Every exhibit tells a story. One shows a piece of metal foil, clear-coated with an air drying or baking synthetic, spotted by nitric acid on the uncoated side. The acid had eaten through the metal only, leaving the transparent film of lacquer unaffected. Steel tubes finished with lacquer withstand hard banging together without chipping, flaking or peeling. On disks finished with flexible finishes, the various stages of blanking and forming, after lacquering are shown.

Other interesting exhibits include an aluminum fan blade, coated with lacquer that had been subjected to 212 deg. F. in a steam oven for two years without loss of flexibility or adhesion, lacquered nameplates and dials, and samples of work illustrating the application of special lacquers before the electro-plating operation, the lacquer withstanding the action of any plating solutions, including chromium, and is buffable without "drag."

A combination finish on zinc and aluminum die-castings was shown to demonstrate the new fashions in finishing, achieved by interesting stencil applications and two-tone work done with only one bake.

The Roxalin Trail Blazer will shortly start a tour to present its story visually and graphically to manufacturers throughout the United States, demonstrating many applications of lacquers that have or can brighten merchandise to make it more salable, no matter how dull or prosaic the base may be, and how lacquer will protect and preserve the product in the ultimate consumers' hands.



***The Roxalin Trail Blazer, finished in blue and aluminum, and a view of its interior showing how the examples of finishes are displayed***

## *Lincoln Announces Contest Rules*

THE general rules and conditions for participation in the arc welding prize contest sponsored by the James F. Lincoln Arc Welding Foundation have been announced. The scope of this contest and the prizes to be awarded was published in the March issue of *Product Engineering*.

With reference to the eligibility of contestants, any person or group of two or more persons may enter the contest but may enter only one paper

on only one subject in any one of the classifications listed. Payment will be made to the person or persons signing the paper. Neither founder, its employees, officers, advertising agency, nor any of its distributing agents shall be permitted to enter the contest or receive any award.

To be eligible each contestant must have actually participated in work upon which the subject matter of his paper is based. The work described in the paper should be the product of the company or firm with which the contestant is or has been connected. The contestant's connection with the company or concern may be either in the capacity of employee or consultant. Consulting engineers may have the work on their product done by a job welding shop.

Other persons, or groups of persons, not so affiliated, may submit a paper on the design of any machine, structure, building, manufactured or fabricated product. The contestant's exact relation to the work must be clearly stated to assure eligibility.

Complete details are given in a book that will be sent free, on request. Address A. F. Davis, secretary, The James F. Lincoln Arc Welding Foundation, P.O. Box 5728, Cleveland, Ohio.

# A.S.M.E. to Meet in Detroit

PLANS for the 1937 semi-annual meeting of the American Society of Mechanical Engineers, to be held in Detroit, Mich., May 17-21, with headquarters at the Hotel Statler, were announced in detail by Sabin Crocker, secretary of the Detroit committee.

Under the chairmanship of Harry T. Woolson, executive engineer of the Chrysler Corporation, the Detroit Papers Committee has outlined a series of six general sessions which will be held on the mornings and evenings of Tuesday, Wednesday and Thursday, May 18-20. On the afternoons of these same days there will be simultaneous sessions and tours to various industrial plants in Detroit. All day Friday will be available for additional inspection tours. Thus it will be possible to participate in these tours without the conflicting interests of technical sessions that might otherwise interfere.

The first general session will be on the subject, "Contribution of Automotive Engineering to Other Fields." E. F. Hirshfeld, chief of research of the Detroit Edison Company, will deliver the opening address. Messrs. Tracy V. Buckwalter and O. J. Horger of The Timken Roller Bearing Company will present a paper on "Modern Locomotive and Railroad Equipment."

Tuesday evening's session will be devoted to "Improved Methods of Fabrication." This session is being sponsored by the Machine Shop Practice Division of the A.S.M.E. At the third general session, scheduled for Wednesday morning and sponsored by the railroad division, Edward G. Budd, president of the Edward G. Budd Manufacturing Company, will speak on the aspects of automotive engineering that have been applicable to railroading. At this session Rupen Eksergian will present a companion paper on the economics of power for light-weight trains.

At the fourth general session, K. T. Keller, president of the Chrysler Corporation, will read a paper on mass production methods, covering such phases as the timing of assembly lines, scheduling material delivery, controlling production to suit dealers' demands and kindred topics. Also on Wednesday evening, W. J. Cameron, well-known radio speaker of the Ford Motor Company, will talk on the decentralization of industry.

At the fifth general session, devoted to steel and its applications, T. F. Olt, supervising metallurgical engineer of the American Rolling Mills Company, will talk on special purpose steels and the development of wide strip mills. Another paper will deal with grain structure of deep drawing steels.

A list of papers and speakers will be published in the May number of *Product Engineering*.

# MEETINGS

**Southern Textile Exposition**—Textile Hall, Greenville, S. C. April 5–10 William G. Sirrine, president, Textile Hall Corp., Greenville, S. C.

**International Association for Testing Materials** — International Congress, London, England, April 19–24. K. Headlam-Morley, 28 Victoria St., London, S.W.1.

**Knitting Arts Exposition**—Commercial Museum, Philadelphia, Pa. April 19–23. A. C. Rau, secretary, 327 Park Square Bldg., Boston, Mass.

**American Foundrymen's Association**—Annual convention and foundry show, Milwaukee, Wis. May 3–7. Arthur J. Tuscany, secretary, 632 Penton Building, Cleveland, Ohio.

**American Society of Mechanical Engineers** — Semi-annual meeting, Hotel Statler, Detroit, Mich. May 17–21.

# Summer Conference on Strength of Materials

A SPECIAL summer school and conference on strength of materials will be held at the Massachusetts Institute of Technology for four weeks beginning June 21. The course will include lectures, laboratory tests, seminars and discussions by members of the M.I.T. faculty and engineers prominent in industry.

The subject of creep will be presented by Dr. A. Nadai, consulting engineer, Westinghouse Electric & Manufacturing Company and also C. R. Soderberg, manager of the Steam Turbine Department, Westinghouse Electric & Manufacturing Company, in which design applications using plasticity theory will be discussed.

Lectures on fatigue will be given by Dr. H. J. Gough, superintendent, Engineering Department, National Physical Laboratory, England.

The lectures on strength of materials will be given by members of the Institute staff. The laboratory exercises in the testing of metals will make use of the more modern measuring instruments and apparatus.

Four seminars will be held during the course to afford opportunity for the presentation of recent developments in allied fields of engineering mechanics.

The special summer course in strength of materials will be concluded by two all-day conferences on fatigue and creep, at which various aspects of these two subjects will be discussed.

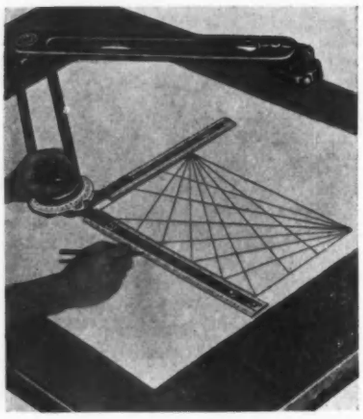
For the complete program the tuition fee will be $80. Due to limitation of laboratory facilities, admission to the school will be in order of application. In any event, all those intending to take part should register by June 1. Further particulars may be had by communicating with Prof. John M. Lessells, Department of Mechanical Engineering, Massachusetts Institute of Technology, who is director of this summer school.

# Do You Know That—

- Oiliness has been defined as a term signifying differences in friction greater than can be accounted for on the basis of viscosity when comparing different lubricants under identical test conditions. (47)

- You can hold between your thumb and index finger what is said to be the smallest internal combustion engine made in regular production. The bore is 0.709 in., and the stroke is 0.788 in. making 0.3 cu.in. displacement. (48)

# New Materials and Parts

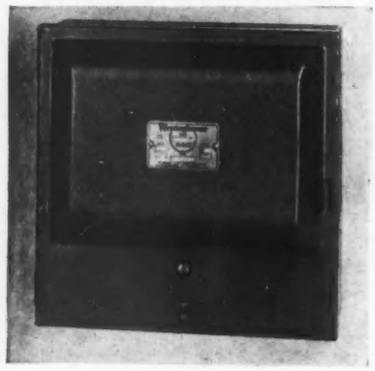


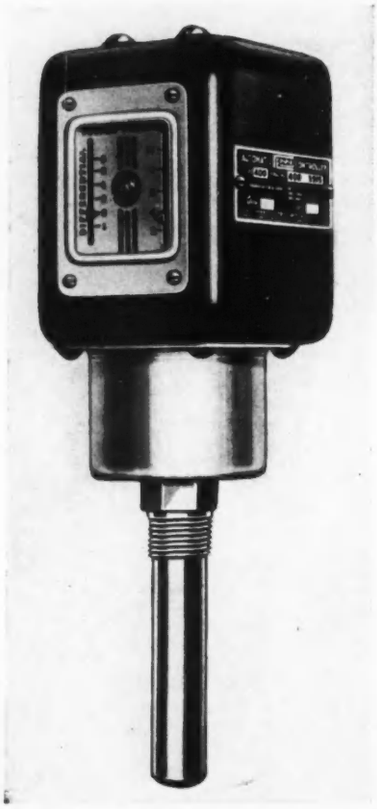
## *Drafting Machine*

Controlled entirely by left hand. Protractor head allows scales to be set and locked at any angle, to move freely in angular displacement, or to stop automatically at every 15 deg. Variations of scale angle are made by means of a locking and stop mechanism operated by lever under left thumb of operator. Parallel motion maintained by tempered steel bands under constant tension which are concealed within arm sections. Uniformity of operation provided by use of precision ball bearings for all moving parts. Available in standard type, with spring counter-balance to prevent accidental motion, for use on boards at angles up to 15 deg., and in vertical type, with weighted counter-balance, for use on boards in any position from horizontal to vertical. Keuffel & Esser Co., Hoboken, N. J.

## *Midget Contactors and Line Starters*

Designed to fit in unusually small spaces, this line of contactors and line starters comprises a group of remote-controlled reversing and non-reversing magnetic starters for small single- and polyphase motors. Operating mechanism is mounted on an insulating base and may be removed as a unit from cabinet. A thermal relay, with inverse-time-limit characteristic, affords overload protection. Requiring only a small control current, the devices may be operated from two- and three-wire thermostats, pyrometers, pressure gauges and other delicate instruments, as well as from pushbuttons. Reversing contactor consists of two two-pole contactors mechanically interlocked to prevent both from closing at same time. Reversing line starter is made up of a reversing contactor and a single-pole overload relay for single-phase motors and a two-pole overload relay for three-phase motors. Recommended for use with motors driving air conditioning equipment, small compressors, small pumps, small fans, domestic stokers, and small machine tools. Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.



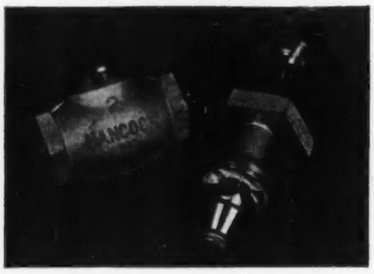


## *Immersion Switch*

Two-wire type, opens circuit at high temperature and closes at low. As a safety limit switch this control shuts down automatic equipment in the event of excessive temperatures on steam, hot water and vapor systems. Contact mechanism is inclosed in dust and tamper-proof case. Can be obtained with both adjustable and differential range, with outside adjustments, and with calibrated scale. Rated to include ½ hp., 110-220 v., a.c.; for 10 amps., 110 v., a.c., non-inductive; or 5 amps., 220 v., a.c., non-inductive service. Penn Electric Switch Co., Des Moines, Iowa.

## *Larger-Size Valves*

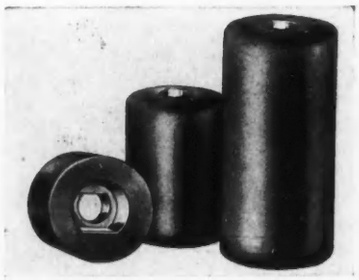
Feature unusually hard stainless-steel seats and disks, heat-treated to 500 Brinell hardness and machined on



diamond boring machine. Hardness claimed to give valve added resistance to wire drawing and steam cutting. New sizes added to previous line of 1 in. and under are 1¼, 1½ and 2 in., in globe and angle patterns. Hancock Valve Div., Consolidated Ashcroft Hancock Co., Bridgeport, Conn.

## *All-Rubber Flexible Couplings*

Made with flat spots molded into end sockets to fit securely over flats on shafts, thus eliminating set screws. Soft, live rubber absorbs shock and vibration. Light in weight, so as not to unbalance small rotors. Flexible

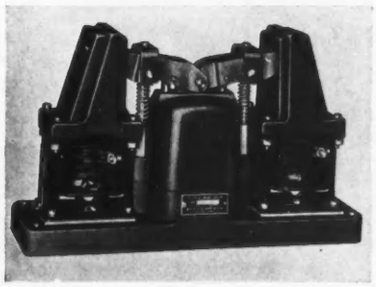


throughout entire length since no metal parts are used. Endwise movement for allowing rotors to find their correct

centers can be controlled by lengths of flats on shafts or by depths of pockets in coupling. Operate quietly, having no parts to vibrate or rattle. Available in a variety of lengths from 1¼ to 4¼ in. and bores from ¼ to ¾ in. Henry Engineering Co., Chicago, Ill.

## *Solenoid Air Valve*

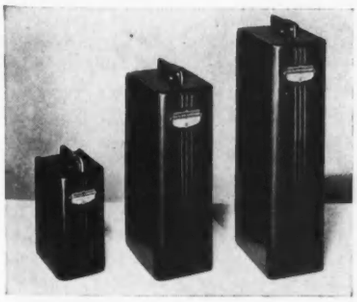
May be operated as six-way or double three-way control valve or as four-way compound exhaust valve. The two quick-acting solenoids operate independently, each having a spring return. Valving units are inherently balanced, operate with short travel, and are so constructed that seal becomes tighter as pressure is increased.

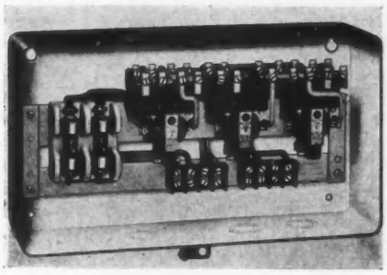


Units have no metal-to-metal wearing surfaces. Stainless steel valve bodies. Cage or solenoids, mounted on plate to which piping is fitted, may be removed for inspection without disturbing piping. Available in ⅜, ½ and ¾ in. sizes, normally closed type. Normally open type to specification in same sizes. C. B. Hunt & Son, Salem, Ohio.

## *Drum Controllers*

Removable case incloses a compact mechanism which offers features such as an insulated cylinder to which segments are secured, renewable contact fingers mounted in individual plastic moldings, accessibility for wiring, and a self-indexing positive-action drum. For use with reversible a.c. squirrel cage motors and reversing or non-reversing multi-speed squirrel-cage motors, driving machine tools and similar machines where speed regulation is desired. Cutler-Hammer, Inc., Milwaukee, Wis.





## *Automatic Alternator Control*

For use with duplex installations in which one unit is held in reserve for peak loads or used as standby in case of breakdown. Automatically switches control circuit from one motor starter to other each time it operates. Can be used with any duplex motor installation, such as in refrigerator units, fans, and compressors, and with any pilot control device, such as a float switch, pressure switch, vacuum switch, thermostat, or two-wire push-button. Allen-Bradley Co., Milwaukee, Wis.

## *Heavy Duty Rollers*

Both designs have self contained double seals that protect ball bearings from dirt and grit. The hexagon axles are provided with grease fittings and are drilled to feed grease through

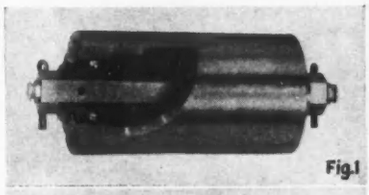


Fig.1

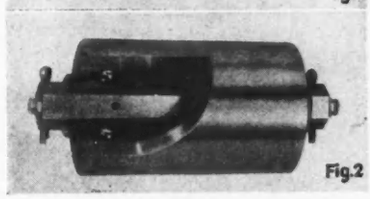
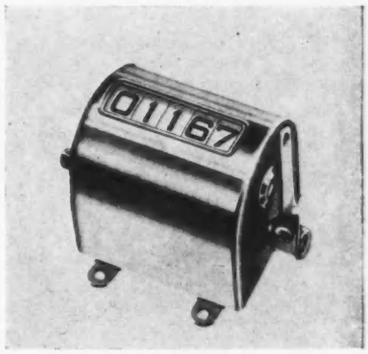


Fig.2

inner ball race. Seamless steel tubing is used for rollers. Roller shown in Fig. 1 is furnished in 4, 4¼ and 4½ in. diam. sizes, and has a carrying capacity from 3,000 to 5,000 lb. Roller shown in Fig. 2 is 5 in. in diam. and has a load rating of 8,000 lb. Frame rails of various shapes and sizes are available. Mathews Conveyor Co., Ellwood City, Pa.

## *Thermosetting Phenolic Resins*

In both liquid and powdered forms for industrial bonding applications. Certain types used with or without rubber and drying oils in impregnating loose or woven asbestos produce high heat-resistance, strength, and a stable coefficient of friction. Other resins are used for bonding ground cork to produce heat-resisting gasket stock. Other forms are used for the production of laminated tubes using the dry resin process. General Plastics, Inc., North Tonawanda, N. Y.

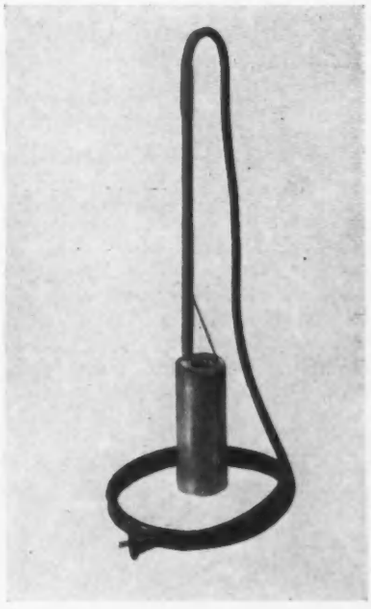



## *Small-Size Counter*

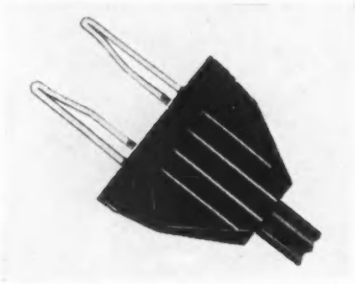
Lever throw of ½-in. in forward direction only gives positive action, bringing new figure into completely legible position. Protection against skips or overthrow of number wheels is provided. Operating arm may be mounted on either right- or left-hand side of device. Cadmium-plated brass case. For dispensing units, production machinery and general application on small machines. Overall dimensions: 1⅞ x 1½ x 1¾ in. Production Instrument Co., Chicago, Ill.

## *Portable Immersion Heater*

Hollow-cylinder shaped element presents largest area of heating surface possible for given length and diameter, and also assists in promoting circula-



tion of liquid being heated. Rubber tubing between metal riser and rubber-covered wire cord prevent liquid from working into heating element in event of accidental over-immersion. Elements available in lengths of from 2 to 8 in. standard, with risers and longer length elements to specification. Standard elements rated from 1,300 to 4,000 watts, either 110 or 230 volts. Edwin L. Wiegand Co., Pittsburgh, Pa.



## *Molded Attachment Plug*

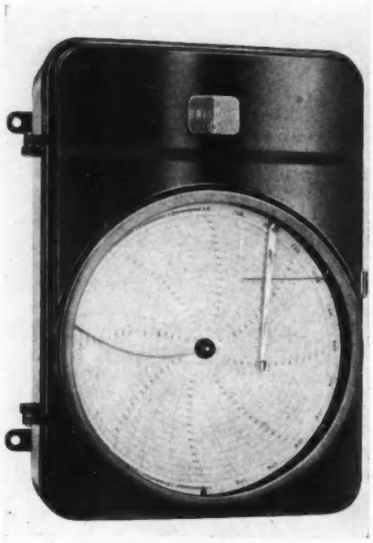
Features spring-action blades, assuring positive contact even in worn sockets, and solid molded-plastic construction designed to permit rough handling. Compact dimensions allow use in new five-way outlets. Blades and wire are arranged to withstand heavy pull and provision is made to prevent twisting of wire from causing short-circuits. Allied Mercantile Co., Irvington, N. J.

## *Flexible Finish*

Produces finish resembling that obtained by extruding or cementing celluloid upon a metal base. Is flexible, durable and applicable to any metal. Known as Duflex it is recommended for high-grade metal products subjected to extremely hard usage, such as golf sticks and fishing rods. Maas & Waldstein Co., Graybar Bldg., New York, N. Y.

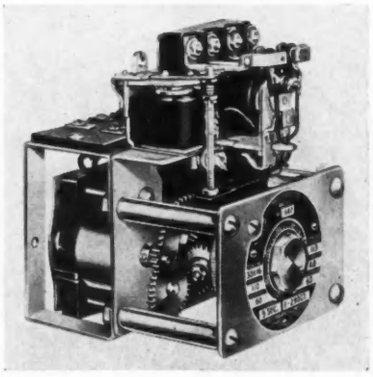
## *Flexible Coupling*

Equipped with insert having hardened and ground balls that roll on hardened and ground jaw surfaces to carry the load. End play of shaft is unrestricted. Holes and hubs are soft permitting alterations. Capacity at 300 r.p.m. is $\frac{1}{16}$ hp. Available in $\frac{1}{2}$, $\frac{5}{8}$ and $\frac{3}{4}$ in. Bores. Boston Gear Works, Inc., North Quincy, Mass.



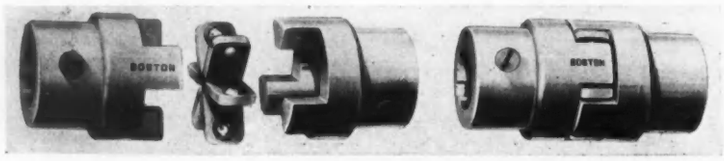
## *Round-Chart Potentiometer*

Pen actuated so as to follow changes in measured quantities as they occur and at rate dependent upon rate of change. No mechanical motion of any kind takes place except when a change in the measured quantity takes place. Is not affected by normal plant vibration or by atmospheres laden with dust, moisture and corrosive fumes. Made up of five compact units which are replaceable and have no mechanical connection between galvanometer unit and other units. Requires no lubrication. Available as a pyrometer, tachometer, resistance thermometer, millivoltmeter, and milliameter. The Bristol Co., Waterbury, Conn.



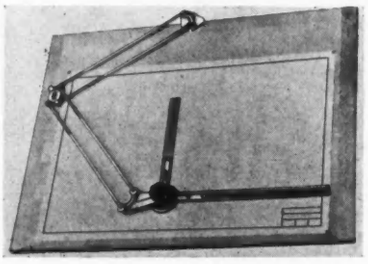
## *Time Delay Relay*

Driven by synchronous motor and provides definite adjustable time interval between two different operations. Differential gearing eliminates the use of clutches. Has a magnetically operated brake. A midget relay is built into the device. At the instant of tripping the contacts a limit switch disconnects the motor. Relay is designed for 110 and 220 volts a.c. operation and is immediately recycling. Quick make and break contacts are furnished with all contact combinations. Ward Leonard Electric Co., Mount Vernon, N. Y.




## *Larger-Size Drafting Machine*

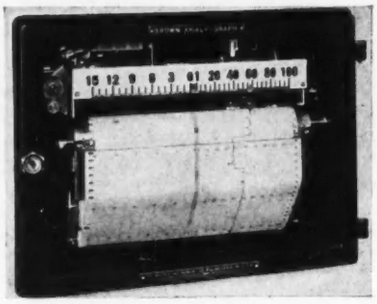
Will make drawings up to 28x38 in. Has permanently calibrated parallel mechanism and is assembled with precision ball bearings prelubricated



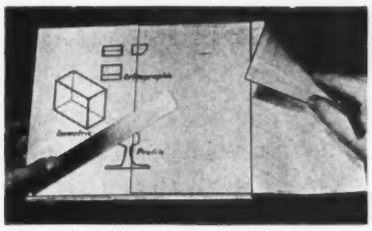
and sealed at factory. New-type protractor used has automatic stop permitting accurate return to zero position. Scales are of Duralumin and chuck-plate adjustment provides for accurate right-angle alignment. Aluminum alloy castings are used for machine and all parts are given a dull, satin finish to prevent glare. Wrigraph Sales Division, 5207-10 Euclid Ave., Cleveland, Ohio.

## *Furnace Atmosphere Quality Recorder*

Operates on the principle of measuring the thermal conductivity of the gas mixture which is continuously drawn from the furnace. It applies poten-



tiometer accuracy to the measurement of furnace atmosphere quality. To render the instrument equally useful for measuring all types of atmospheres, the chart has been divided into two equal sections, one section is used when combustion is complete and the other is used when combustion is incomplete. The large scale permits observation at a distance. The Brown Instrument Co., Philadelphia, Pa.



## *Sketch Pad Set*

Consists of a tracing paper pad sealed on three edges, a guide plate and a steel straight-edge. The buff colored guide plate, printed with an isometric design in blue lines on one side, slides under the top sheet of the pad. The other side is printed 8x8 lines to the inch with every ½ in. line heavy. The guide plate is moisture, dirt and crack proof, and is easily cleaned. Made in pocket size, 200 sheets, 5½x8½ in.; brief case size, 100 sheets, 8½x11 in.; and office size, 50 sheets, 11x17 in. Wade Instrument Co., 2246 Brooklyn Station, Cleveland, Ohio.

## *Electric Counter*

Has indicating wheels geared so that load on operating mechanism is constant regardless of number being indicated. Purpose is to eliminate tendency of counter to skip when only one wheel should be turning and miss when all wheels should be turning. Duty of operating mechanism is said to be practically the same as that of a relay. Available with five digits, in either reset or non-reset type. Struthers Dunn, Inc., Philadelphia, Pa.

## *Compensating Welding Rod*

Evenly tapered so that it is small at starting end and more material is deposited as the rod is used in order to offset the increased heating effect as welding progresses. Also compensates for electric-current heating effect in that part of rod not yet deposited. Claimed to require less power, especially in starting arc, when rod is used for corner and deep V welding. Available in bare, fluxed, dipped and extruded electrodes in all standard sizes. Electric Arc Cutting & Welding Co., Newark, N. J.

# MANUFACTURERS' PUBLICATIONS

**Bearings** — Merriman Bros., Inc., 185 Amory St., Jamaica Plain, Boston, Mass. 12 pages, 8½x11 in. Description, engineering data on load carrying capacities, speeds, efficiency, etc., and applications of Lubrite self-lubricating bearings and bushings, including applications of high temperature bearings.

**Controlled Lubrication** — Alemite Div., Stewart-Warner Corp., 1826 Diversey Parkway, Chicago, Ill. Catalog, 56 pages, 8½x11 in. Presents the entire range of Alemite industrial equipment—hand guns, power-operated units, fittings, valves, meters, etc., well illustrated with photographs and drawings. Numerous photographs of the equipment in actual use in various industries are included.

**Die-Casting**—The New Jersey Zinc Co., 160 Front St., New York, N. Y. 32 pages, 8½x11 in. A pictorial presentation "Supplement to a Visual Report of Progress" representing the progress of zinc alloy die-castings in the past year.

**Enameler's Dictionary**—Ferro Enamel Corp., Cleveland, Ohio. Booklet, 3½x5 in., 80 pages. Defines terms and names used by enamelers for finishes and materials.

**Graphite Bronze Bearings**—Randall Graphite Products Corp., 609-613 W. Lake St., Chicago, Ill. 24 pages, 4¾x7 in. Lists inside and outside diameters and lengths of standard bronze graphite bushings and sheave bearings. Also gives figures on inside diameter reductions for pressed fitted bearings. A feature of this catalog is its marginal thumb index for quick reference.

**Leather Products for Industrial Purposes** — Alexander Brothers, Inc., Philadelphia, Pa. 8½x11 in., 32 pages. Presenting information on belting and packings, includes tables of power ratings and speeds for belts.

**Machine Lamps**—Fostoria Pressed Steel Corp., Fortoria, Ohio. Bulletin, 12 pages, 8½x11 in. Illustrating types of lighting units for machines and industrial uses.

**Making Commutators** — Reliance Electric & Engineering Co., Cleveland, Ohio. Folder 8½x11 in. Shows methods used for manufacturing, testing and inspecting commutators.

**Nickel Alloy Steels**—The International Nickel Co., Inc., 67 Wall St., New York, N. Y. Bulletin U-2, 12 pages, 8½x11 in. Data and tables on the application of nickel alloy steels in petroleum production equipment.

**Nickel Silver Alloys**—The Riverside Metal Co., Riverside, N. J. Revised edition. 48 pages, 8½x11 in. Reference booklet covering the various nickel silver alloys and the forms in which they are available, as produced by Riverside. Well illustrated with examples of products made of these alloys. Characteristics and uses are explained in detail. Tables of physical properties, electrical resistance, standard commercial tolerances and many others are included.

**Oil Testing Instruments**—C. J. Tagliabue Mfg. Co., Park & Nostrand Aves., Brooklyn, N. Y. Catalog No. 699D, 24 pages, 8½x11 in. Lists and describes complete line of oil testing instruments, hydrometers and thermometers. Contains an official guide to the principal tests of petroleum products.

**Porcelain Enameling**—The Horace R. Whittier Co., Pequabuck, Conn. 5¾x9 in., 16 pages. Tells the story of Whittier porcelain enameling with illustrations of processes used in manufacture.

**Right Angle Speed Reducers**—D. O. James Mfg. Co., 1120 W. Monroe St., Chicago, Ill. Three bulletins each 8½x11 in. Bulletin No. 11, 48 pages, describes, illustrates and gives dimensions of spiral bevel gear speed reducers, with ratios ranging between 1 to 1 and 6 to 1 inclusive. Bulletin No. 16, 20 pages covers spiral bevel gear reduction units combined with a herringbone gear pair for ratios between 6 to 1 and 45 to 1 inclusive. Catalog No. 141 covers bevel gear units combined with a planetary gear system to form a reducer unit, ratios between 8 to 1 and 1200 to 1 inclusive.

**Safety Starting Switches**—The Arrow-Hart & Hegeman Electric Co., Hartford, Conn. Catalog No. 7, 9x10½ in., 58 pages. Contains tables, specifications and illustrations covering safety switches, service equipment, manual and magnetic motor controllers, master devices and circuit breakers.

**Screws** — The Holo-Krome Screw Corp., Hartford, Conn. Catalog 20, 38 pages, 8½x10¾ in. Detailed description covering Fibro forged screws with numerous pictures, drawings and tables.

**Speed Reducers** — Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa. Publication (D.D. 25-030), 8 pages, 8½x11 in. Data on design, efficiency and ratings of types SH and DH speed reducers, single and double reduction series of the single helical gear type.

**Temperature Controllers**—C. J. Tagliabue Mfg. Co., Park & Nostrand Aves., Brooklyn, N. Y. Bulletin No. 1142, 4 pages, 8½x11 in. Describing latest model self-operating controller for temperature and pressure.

**Vibrating Screens** — Productive Equipment Corp., 4600 S. Kedzie Ave., Chicago, Ill. 32 pages, 8½x11 in. Data on selective throw eight position adjustment vibrating screens, including dimension drawings and tables and a list of materials handled.

**Zinc Metals and Alloys**—The New Jersey Zinc Co., 160 Front St., New York, N. Y. 27 pages, 3¾x7 in. Information on all metal products of this company with the exception of rolled zinc. A large section is devoted to Zamak-zinc die-casting alloys with figures on physical properties, corrosion resistance, A.S.T.M. specifications, etc.

# BOOKS AND BULLETINS

## *Gear Design Simplified*

*Franklin D. Jones.* 134 pages, 8½x 11 in. Blue clothboard covers, 201 illustrations. Published by The Industrial Press, 148 Lafayette St., New York, N.Y. Price $3.

The book consists exclusively of formulas, working rules and data actually required by the designer in calculating various types of gears. The theoretical side of gear design has been excluded for simplification. Types of gears dealt with include spur, internal, straight and spiral bevel, helical gears for both parallel and angular drives, double-helical or herringbone and worm gears.

There are problems for each type of gear, each problem being illustrated with a line drawing or diagram. Whenever any problem has more than one solution or angle of aproach, the different solutions are given with the rules, formulas and worked-out examples for each instance. Speeds and ratios, including those for planetary transmissions, and power-transmitting capacities for the different types of gears are given. Not only are the various full-depth standards and stub tooth systems explained and worked out, but the standard module system, employed in countries using the metric system, is included. There is also a table of standard DIN modules with equivalent diametral and circular pitches as well as the important tooth dimensions.

## *Art and the Machine*

*Sheldon Cheney* and *Martha Candler Cheney.* 307 pages, 7x9 in. Numerous illustrations. Clothboard covers. Published by Whittlesey House, McGraw-Hill Book Co., Inc., 330 West 42d St., New York, N.Y. Price $3.75.

Principles of utility, economics and art, all have been merged and blended by present day industrial designers. A new concept and method of procedure is making itself manifest. It has changed the appearance of a great variety of common machine-made articles which perform all sorts of services; it is commonly known as modern design.

The authors present in this volume an interesting non-technical discussion of the beginnings of the art-expression movement, its purposes, evolution of trends, and effect on contemporary mass production of goods. Many illustrations are used showing the effect of the artist's technique on merchandise, appliances, machinery and architecture.

While intended for the general reader, the book contains much that is of interest to designers and industrialists that is thought provoking and inspiring.

## *The Mechanical Testing of Metal and Alloys*

*Field Foster.* 253 pages, 5½x8½ in. Red clothboard covers. Published by Pitman Publishing Corp., 2 West 45th St., New York, N. Y. Price $3.75.

In addition to covering descriptions of modern testing equipment and their operation, this book includes a brief review of the elementary theory of elasticity and the structure of metals. Not only does the author describe the construction and operation of the various types of physical testing machines in commercial use, but he also explains how the results of the tests are recorded and interpreted.

Chapters include universal testing machines, tension and bending tests, testing machine accessories, extensometers and recorders, torsion testing machines, hardness testing machines, notched-bar impact testing, fatigue testing machines, and wire testing machines. An interesting chapter entitled "Some Test Phenomena and Results" discusses types of fractures, hysteresis, types of failures, creep tests and similar things. A comprehensive bibliography lists about 45 books and articles, many of which are referred to in the text.

## *Materials Handbook*

*George S. Brady.* 634 pages, 4x7 in. Semi-flexible covers. Published by McGraw-Hill Book Co., Inc., 330 West 42d St., New York, N. Y. Price, $5.

As explained by the author in his preface, this book is not a metallurgical treatise, but is intended for use as a reference guide for the selection of engineering materials. The various materials are listed in alphabetical order, both generic terms and trade names. The name of the maker is given in every instance where the alloy is limited to one producer. Metals, nonmetals, the chief ores and the most important industrial chemicals are included. Patented and trade-name materials are in many cases grouped in their respective classifications in order to give a broader picture of particular subjects. Many new materials have been added in this third edition.

For each material is given its origin or method of manufacture, a brief description of its general properties, its chemical composition and principal physical properties and uses.

### *The Art and Science of Spring Making*

*Barnes-Gibson-Raymond Div. of Associated Spring Corp.,* 6400 Miller Ave., Detroit, Mich. 88 pages, 8½x11 in.

A practical handbook, well presented and illustrated, on the selection and design of all forms of springs, including compression springs, extension springs, valve springs, flat springs and Belleville springs. Tables, formulas, calculations and complete information on the characteristics of the various springs are presented and numerous problems in design actually worked out for easier understanding.

Written in a not-too-technical language, the book is probably the most complete up-to-date discussion on spring making available and should prove of value to the user as well as the designer of springs.

### *How to Weld 29 Metals*

*Charles H. Jennings,* Engineer in Charge of Welding Research, Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa. 100 pages, 8¼x5¼ in. Price, 50 cents.

A practical and usable treatise on electric arc-welding covering the procedure, conditions and materials for welding modern alloys. Includes data for welding all types of joints with varying thicknesses of metal, such as electrode diameter, welding current, speeds, deposition, etc. The book should prove valuable in simplifying and improving the welding of present day metals and alloys.

### *Contour Sawing*

*Continental Machine Specialties, Inc.,* Minneapolis, Minn. 110 pages, 6¼x9¼ in. Paper covers. Price $1.00.

A handbook describing the process of metal cutting known as contour sawing. The book compares the advantages of contour sawing with various other cutting processes such as shaper work, lathe work and torch cutting.

A section is devoted to shop practice and methods for making punches and dies, jigs and fixtures, templates, cams and special parts. Other sections include miscellaneous applications and contour filing.

### *Tinplate and Tin Can Manufacture in the United States*

*International Tin Research and Development Council.* L. J. Tavener, U. S. Representative, 149 Broadway, New York, N. Y. Bulletin No. 4. 144 pages, 10⅛x8¼ in. 159 illustrations. Paper covers.

Apart from detailed descriptions of methods and machinery used in making tinplate and tin cans, there are chapters devoted to the field for tinplate cans and containers, the history of the industries, statistics, tinplate lithographing, and the future prospects for improvements and new applications. A feature of the bulletin is the large number of photographs and diagrams to illustrate the text.

Tribute is also paid to the benefits resulting from the continual collaboration of the tinplate makers and the users in solving difficulties and carrying on research.

# *Temperature—Resistivity Curves of Electrical Resistors—I*

LEO G. HALL

THE DATA on electrical resistivity as presented here refer to materials of commercial purity, not chemically pure. Because resistivity of all materials varies somewhat, depending on their purity, the values given can be taken as fair averages only and not as exact. The data have been taken from the work of several hundred investigators and in cases of doubt or of conflicting information, check measurements have been made.

Resistivity values shown in the accompanying charts are in ohms per cubic inch. To obtain the resistance of any particular specimen, the value from the chart is multiplied by the length of the specimen in inches and then the product is divided by the cross section area of the specimen in square inches.

There are two general types of curves. The metals in general show increased resistivity with increase of temperature (electronic conduction), while the non-metals show decreased resistivity with rise of temperature (electrolytic conduction). All materials become good conductors at sufficiently high temperature. Metals show a sudden increase of resistivity at their melting point, while non-metals show a sudden drop of resistivity in melting. With few exceptions the effect of impurities on the metals is to increase their resistivity to a much greater degree when cold than when the metals are hot, thereby flattening the curve. Alloys in general have flatter curves than pure metals; those of Nichrome and Kanthal being almost horizontal. The effect of alloying elements on the resistivity of 0.2 per cent carbon steel at room temperature is as follows:

| | | |
|---|---|---|
| Unalloyed | 0.2 carbon steel | 0.0000043 ohms |
| Alloyed with | 5.0 aluminum | 0.0000250 ohms |
| Alloyed with | 5.0 chromium | 0.0000138 ohms |
| Alloyed with | 13.5 chromium | 0.0000234 ohms |
| Alloyed with | 10.0 nickel | 0.0000114 ohms |
| Alloyed with | 6.0 manganese | 0.0000108 ohms |
| Alloyed with | 5.0 tungsten | 0.0000079 ohms |
| Alloyed with | 2.5 titanium | 0.0000098 ohms |
| Alloyed with | 4.0 silicon | 0.0000244 ohms |
| Alloyed with | 5.0 vanadium | 0.0000480 ohms |

*Note:* The last item is based on 1.1 per cent carbon.

Although these alloying elements increase the resistivity at low temperatures by several hundred per cent the resistivities at 1,800 deg. F. are in no case appreciably greater than that of unalloyed steel, and in some cases are less. Data for the complete curves are lacking, only comparative measurements having been taken at elevated temperatures.

The resistivity of all metals when cold is sharply increased by cold working, but subsequent annealing restores the former value. Heat-treatment for hardening or for grain refinement increases the resistivity, while prolonged heating in the region of grain growth decreases it. In general, all materials show decreased resistivity as the structure becomes coarser.

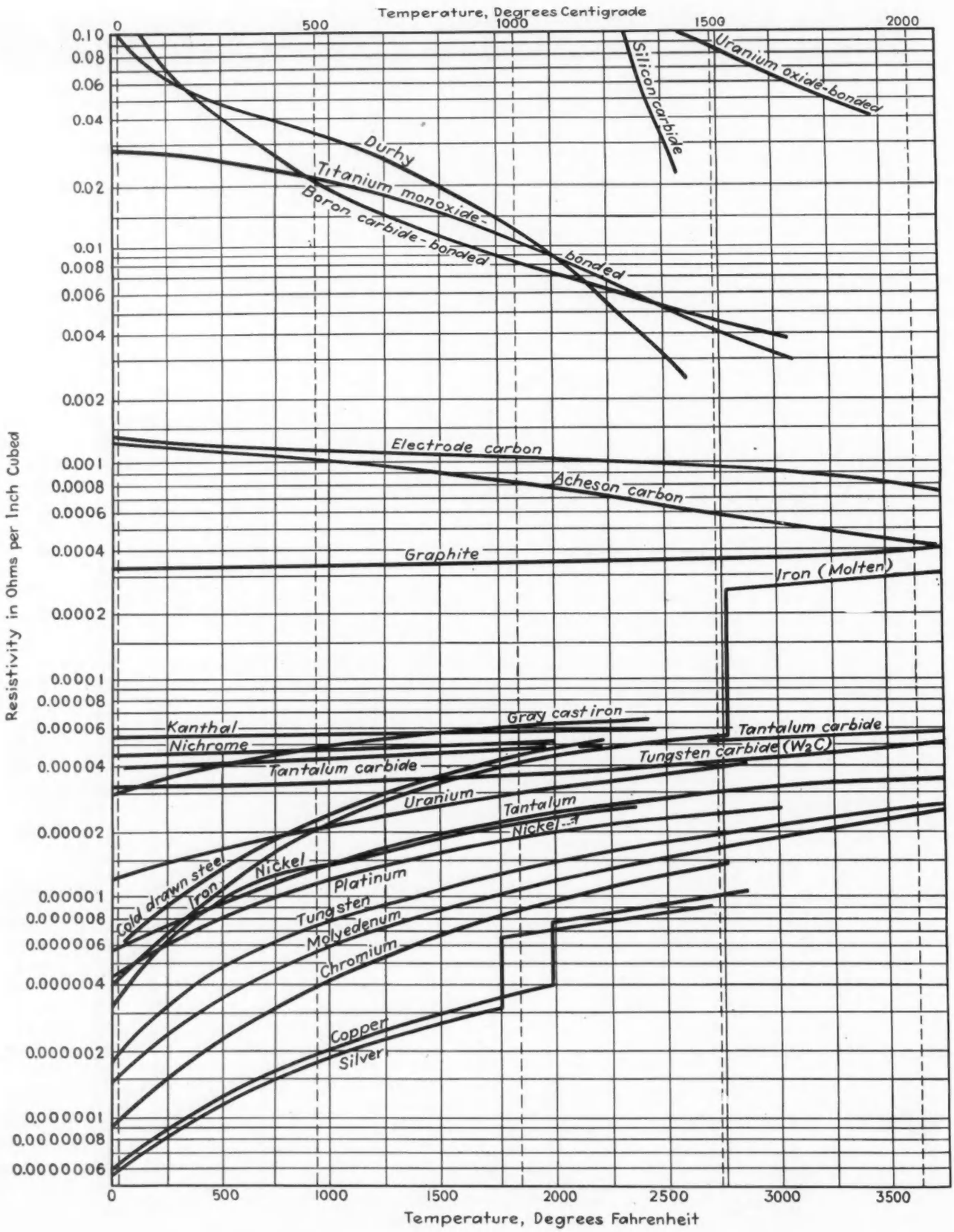
The data on silver, copper, molybdenum, tungsten, platinum, nickel, tantalum, Nichrome and Kanthal apply to the commercial wires. There are three grades each of Nichrome and Kanthal. The curves of the three grades in each case lie so closely together and are so closely parallel that only one of them could be plotted without confusion. The intermediate curves were plotted. The curve for chromium applies to the aluminothermic material, which contains some small percentage of aluminum and is of very coarse crystal structure. That of uranium applies to a specimen containing small quantities of calcium and of iron—probably about 1 per cent total inpurity.

Unless otherwise noted, the curves for the metallic carbides apply to the massive materials. Those marked "sintered powder" were taken from powdered specimens sintered together under considerable pressure so as to form coherent but porous bodies.

The curves for boron carbide, titanium monoxide and uranium oxide were all obtained from specimens of the powders bonded with a small percentage of bentonite, pressure molded and baked to a sufficient temperature to destroy the structure of the bentonite. Uranium oxide is a fairly good conductor at room temperature, is very refractory, and does not burn at high temperatures. Mixtures of it with other materials give promise of improving the characteristics still further, but data are still too fragmentary to be of value.

The curve for graphite applies to small soft graphite electrodes. The larger shapes, being molded under lower unit pressure, are less dense and have higher resistivity. The two curves for carbon probably apply to carbons of different degrees of graphitization, no other explanation being apparent. Both apply to arc electrodes but the source of one of them is unknown.

*Note:*—Curves for non-metals will be published in the May number.

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